United States Patent [19]

Steiger

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[54] **DRIVE FOR AN OSCILLATORY MECHANICAL SYSTEM**

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[73] Assignee: **Sulzer Brothers Limited, Winterthur, Switzerland**

[21] Appl. No.: **591,554**

[22] Filed: **Mar. 20, 1984**

[30] **Foreign Application Priority Data**

Mar. 30, 1983 [CH] Switzerland 1771/83

[51] **Int. Cl.**^{3} **F15B 13/044; F16K 31/56**

[52] **U.S. Cl.** .... **137/625.65; 74/100 R;** 251/75; 251/280; 251/129.09; 251/129.20

[58] **Field of Search** .... 74/97, 100 R, 100 P, 74/106; 137/625.65; 251/75, 137, 138, 280

[56] **References Cited**

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3,095,901 7/1963 Larson et al. .... 251/280 X

Primary Examiner—Gerald A. Michalsky

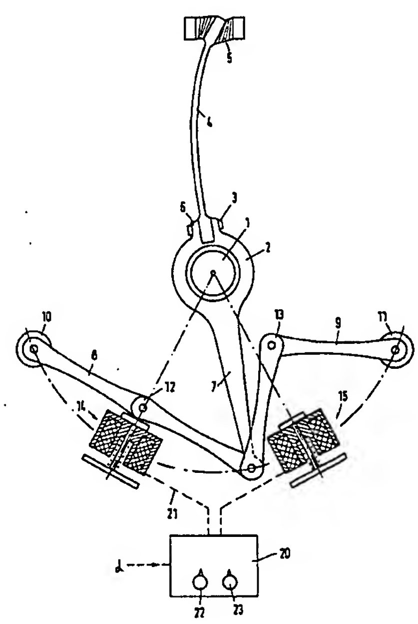
Attorney, Agent, or Firm—Kenyon & Kenyon

[57] **ABSTRACT**

A resiliently prestressed oscillatory system having two discrete operative positions is located by means of toggle levers in the limit positions of the oscillation. One of the toggle levers is in the extended state in the associated operative position and is deflected by a control device through the agency of a movable abutment until the self-locking of the toggle lever is overcome and a prestressed spring drives the system into its other end position.

The toggle levers form an amplifier reducing, for instance, by a hundred-fold to a thousand-fold, the forces which the control device is required to provide. The system can therefore be controlled directly by means of an electronic control without the need for additional pneumatic of hydraulic auxiliary systems or high power electrical systems.

8 Claims, 3 Drawing Figures



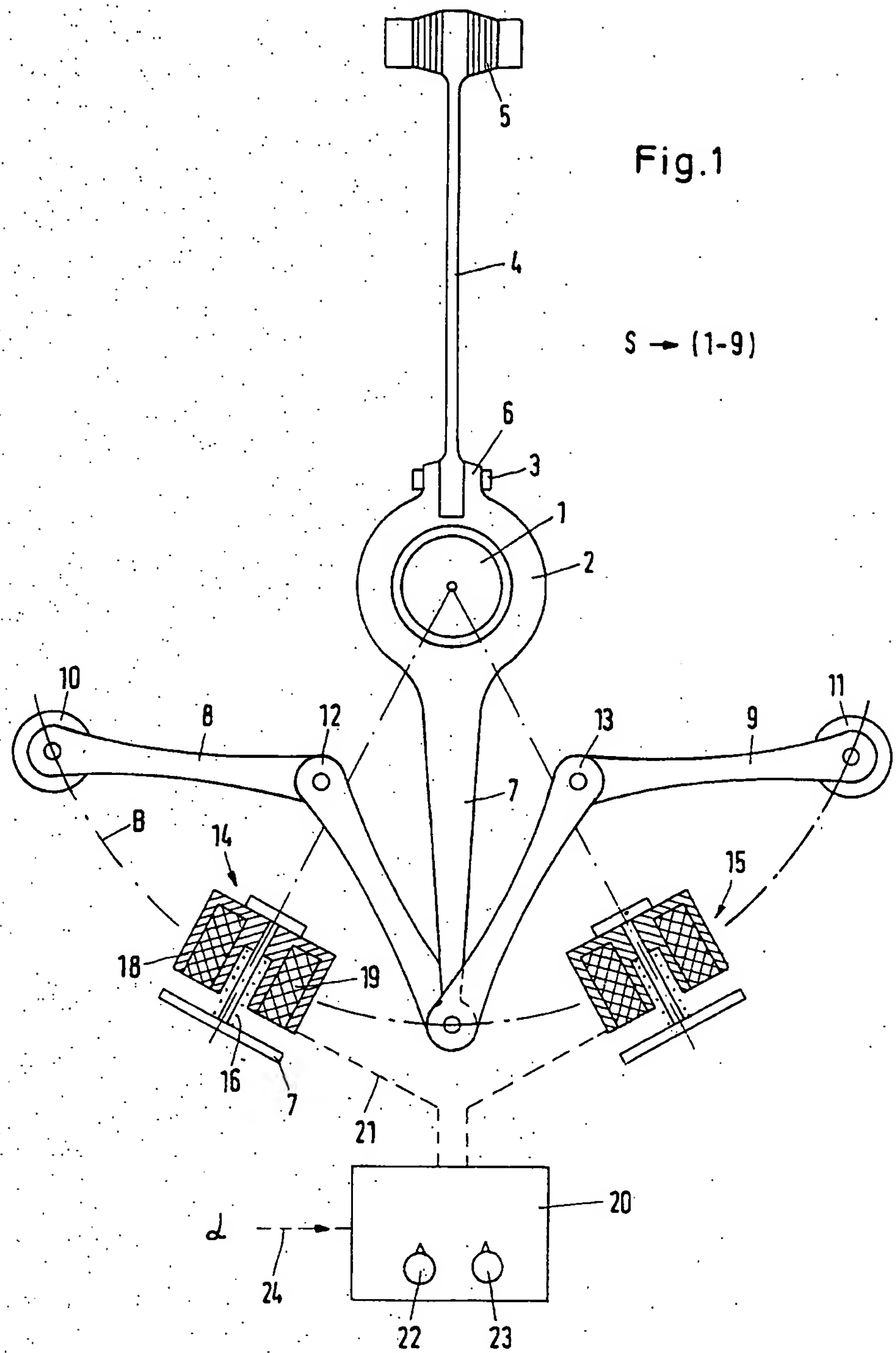


Fig.1

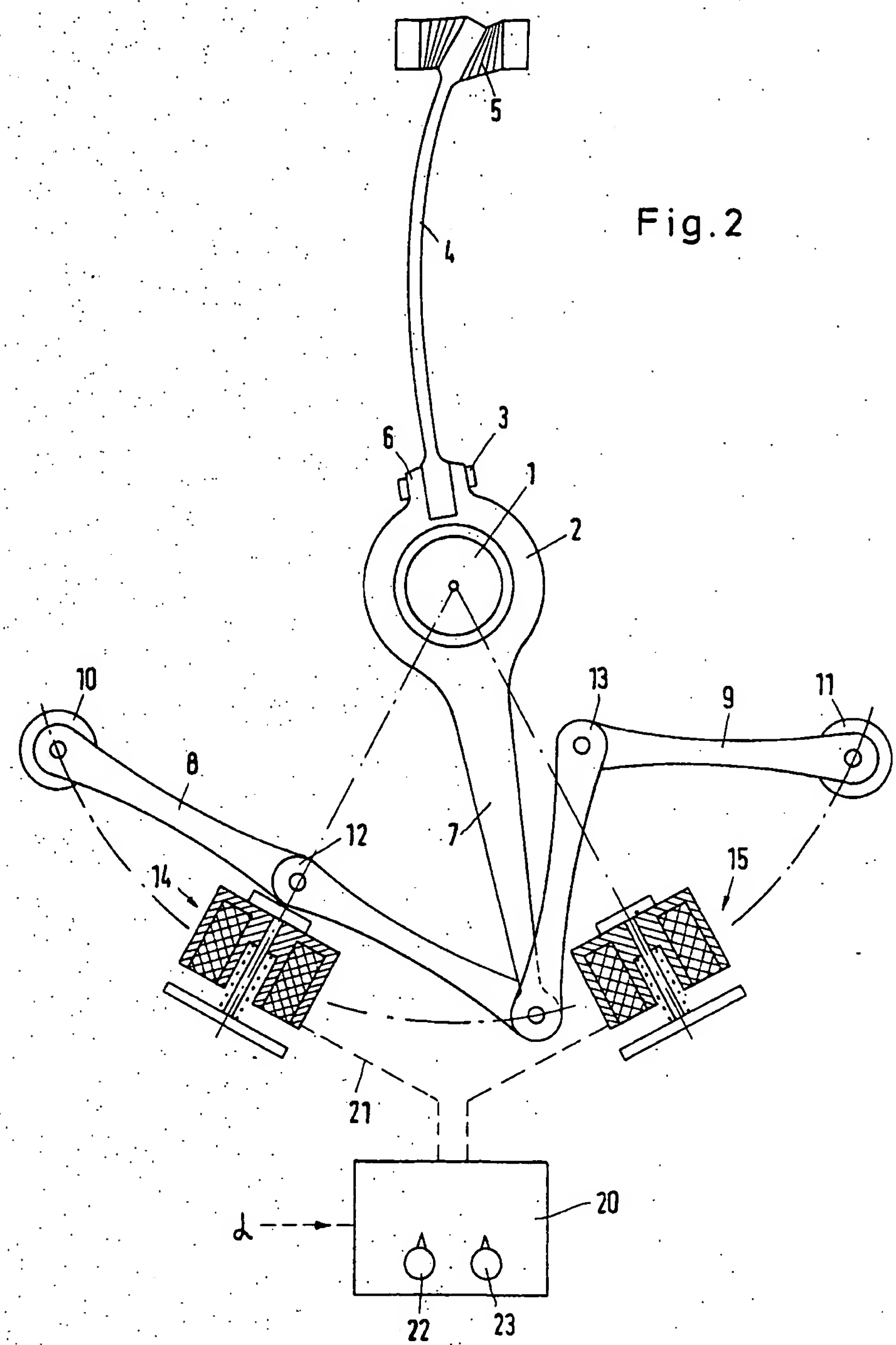
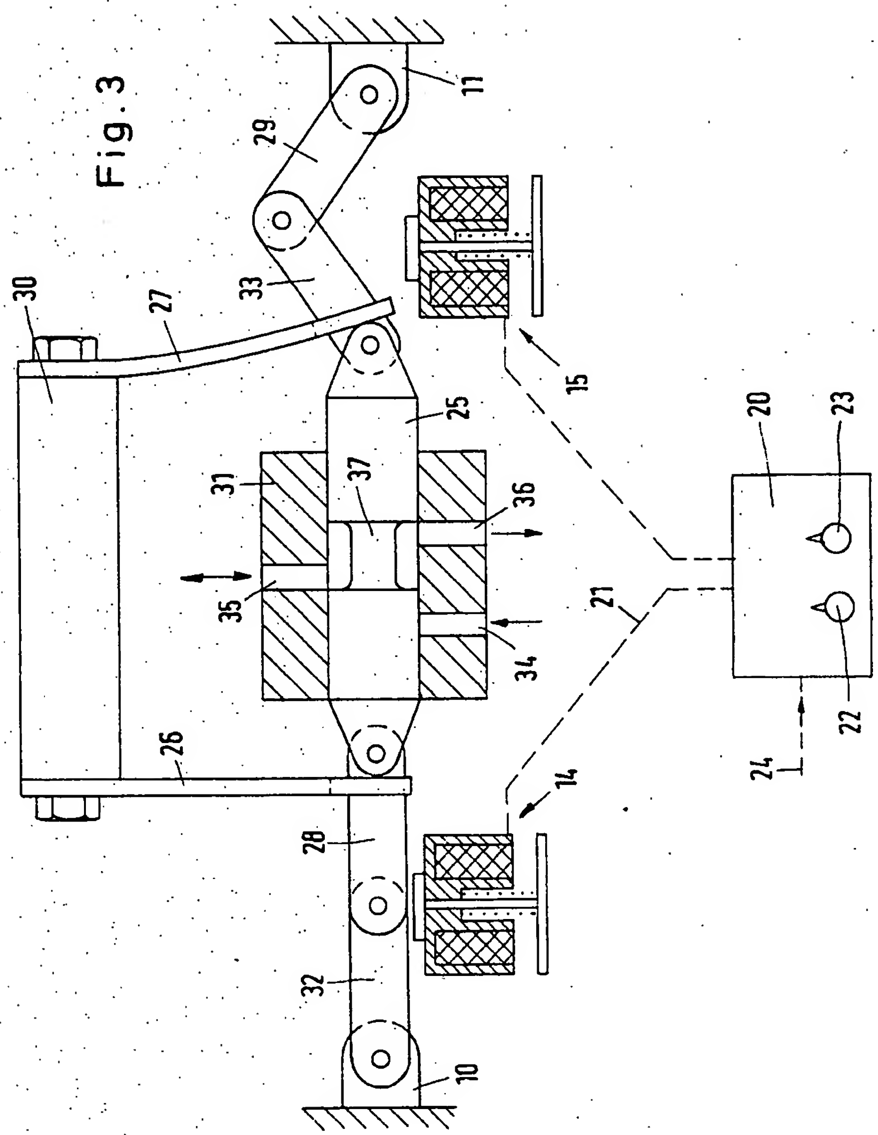


Fig. 2

Fig. 3



DRIVE FOR AN OSCILLATORY MECHANICAL SYSTEM

This invention relates to an oscillating mechanical system and more particularly to a drive for an oscillating mechanical system.

Heretofore, systems have been known wherein the end positions of an oscillating body are exactly defined by the system to be controlled. Examples of such systems include final control elements which move onto a fixed abutment and valve lids which seat on valve seats. Systems have also been known, such as sliding spool valves or rotary slide valves, as described in German O.S. No. 28 07 533 and German P.S. No. 30 16 823, wherein a valve body is located accurately in an end position not by the system to be controlled but by a controlling or actuating mechanism.

Drives have also been known for actuating oscillatory systems of the type wherein one operative position is an end position defined by the system to be controlled, i.e. a valve seat as described in WO-Publication (PCT-Application) No. 81/01626 and German O.S. No. 31 39 399. In these known systems, the forces arising at the elements to be controlled, e.g. at the valves, act directly on the actuating drive which has the function of retaining the element in one of two operative positions. Consequently, these known drives usually have either additional systems, such as hydraulic or pneumatic systems, or high power electromagnets or solenoids whose magnetic field can provide the necessary forces. However, even when the magnetic fields are relatively substantial, the use of direct-acting drives of this kind which have no additional system are limited to relatively small mechanical systems.

In cases where electronic means are used to produce control signals for triggering the movement of a system in either direction, the forces produced by the electronic means by way of magnetic fields is totally insufficient to retain an oscillatory system in an operative position. Hence, additional systems must be provided in order to produce the necessary retaining forces. In these cases, an electronic control means is used only to trigger the additional systems. Consequently, drives of this kind are very expensive and are liable to disturbances.

Accordingly, it is an object of the invention to reduce the forces acting on a drive for actuating an oscillatory mechanical system of relatively large mass with a minimum of energy.

It is another object of the invention to provide a drive for actuating an oscillating mechanical system without the need for auxiliary systems.

It is another object of the invention to have the magnetic forces produced by an electronic control suffice to retain a mechanical system in each of two discrete end positions and to trigger movement of the mechanical system in the event of a changeover.

Briefly, the invention provides a drive for actuating an oscillatory mechanical system which includes a first means for oscillating between two discrete operative positions and a resilient prestressing means for biasing the first means from one position to the other position.

The drive includes a pair of toggle levers which are connected to the first means so as to locate this first means in a respective operative position corresponding to an extended self-locking position of the respective toggle lever. In addition, the drive includes a pair of movable abutments each of which is disposed in the path of a respective toggle lever to inhibit overshooting of the toggle lever past the extended position thereof. Further, the drive includes a control mechanism for selectively moving the abutments to effect a deflection of a respective toggle lever from the extended self-locking position thereof.

The abutments serve to supply a minimum energy to the oscillatory mechanical system so as to compensate for the energy losses of the system.

Each toggle lever represents a mechanical force amplifier which can readily provide "amplifications" of a hundred-fold up to a thousand-fold.

The movable abutments may be of a type which can be energized by electronic signals from the control mechanism. When energized, the abutments deflect the toggles over a relatively long distance so that their reduced forces are sufficient to supply the oscillatory mechanical system with energy to make up the energy losses. The path of the abutments, i.e. the deflection of the "toggle joints" from their extended position, are such that self-locking of the toggle joint is overcome. In this respect, self-locking is due mainly to the friction forces arising in a movement of the toggle levers.

In one embodiment, the oscillatory mechanical system may employ a rotary spindle to which a two-armed lever is secured as the oscillating means. In this case, one arm of the lever is pivotally connected to the pair of toggle levers at a common point. In additition, the prestressing means may include a flexible spring which is secured at one end to a second arm of the two-armed lever while the opposite end is secured to a resiliently deformable mount.

In another embodiment, the oscillating means may include a sliding spool valve having a casing with an inlet bore, a control bore and a discharge bore as well as a spool which is slidably mounted in the casing to selectively communicate the control bore with one of the inlet bore and outlet bore. In this case, the spool is connected at opposite ends to a respective toggle lever while the prestressing means includes a pair of spring strips, each of which is disposed at a respective end of the spool to bias the spool into the casing.

These and other objects and advantages of the invention will become more apparent from the following detailed description taken in conjunction with the accompanying drawings wherein:

FIG. 1 illustrates a diagrammatic view of a drive for a rotary slide valve in accordance with the invention;

FIG. 2 illustrates the drive of FIG. 1 in one operative position; and

FIG. 3 diagrammatically illustrates a drive for a sliding spool valve in accordance with the invention.

Referring to FIG. 1, an oscillatory mechanical system such as an oscillatory rotary slide valve (not shown), for instance, of the kind disclosed in German P.S. No. 3016823, is disposed on a means such as a central spindle 1 for oscillating between two discrete operative positions. As previously stated, a rotary slide valve of this kind has no definitive end position determined by the system to be controlled.

In addition, a two-armed lever 2 is rigidly secured to the spindle 1.

In addition, a resilient prestressing means is provided for biasing the spindle 1 and lever 2 from one operative position to the other operative position. This prestressing means includes a flexible spring 4, such as a leaf spring, which is secured at one end to one lever arm 6 in a pivotal manner by way of a pivot 3. The opposite

end of the spring 4 is secured to a resiliently deformable mount 5.

The drive for the oscillatory mechanical system includes a pair of toggle levers 8, 9 which are pivotally connected to the end of the other lever arm 7 at a common point. Each toggle lever 8, 9 is rotatably mounted on fixed points 10, 11 which lie on an arc B which the other toggle ends, together with the end of the arm 7, describe when rotating about the spindle 1. As indicated, each toggle lever 8, 9 has a joint 12, 13 at an intermediate point.

When the oscillatory system S comprising the components 1,2,4,5, and 7 is in an operative position or end position, the spring 4 is in a stressed state as shown in FIG. 2. This state is produced, for example, by a stressing tool (not shown). In this state, the arm 7 has been deflected from the neutral position shown in FIG. 1 so that one of the toggle levers (8) is extended while the other toggle lever (9) is bent at the joint 13 to the maximum possible angle. In the extended position, the toggle lever (8) is self-locking.

The drive also includes a pair of movable abutments 14, 15 each of which is disposed in the path of a respective toggle lever 8, 9 to inhibit overshooting of the toggle lever past the extended position thereof. In the example shown, the abutment 14 includes an electromagnet or solenoid having an armature 17 which is movable against the force of a spring 16 within a core 18 in which a winding 19 is disposed. The abutment 15 for the toggle lever 9 is constructed in a similar manner. The two abutments 14, 15 are so positioned as to cooperate with the toggle joints 12, 13.

The drive also includes a control mechanism 20 for selectively moving the abutments 14, 15, i.e. the armatures 17 of the respective abutments in order to effect a deflection of a toggle lever 8, 9 from an extended self-locking position as indicated in FIG. 2. This control mechanism 20 is in the form of an electronic control which is able to emit a pulse signal to the winding 19 of a selected abutment 14, 15 via a suitable signal line 21. The electronic control 20 has an independent set-value adjustment 22, 23 for the left-hand and right-hand abutments 14, 15, respectively, and receives an input signal by way of a signal line 24, for example, a signal representing the crank angle α of an internal combustion engine.

In use, the oscillatory mechanical system S operates at a predetermined natural frequency so that the transit time from one end position or operative position to the other is fixed. Consequently, all that can be varied is the beginning of the movement in either direction, i.e. the time of triggering a movement of the right-hand or left-hand toggle lever 8 or 9 or of the abutment 14 or 15. This triggering time can be adjusted independently for each abutment 14, 15 by means of the set-value adjustment 22, 23.

At the initial assembly or after servicing or repairs, a tensioning tool (not shown) is used to move the lever 2 against the force of the spring 4, for example, into the right-hand limit position illustrated in FIG. 2. In this position, the left-hand toggle lever 8 blocks the system S with the armature 16 of the left-hand abutment 14 defining the extended position of the toggle lever 8.

When the control device 20 receives an input signal by way of the line 24 to output a signal to trigger the left-hand abutment 14, the armature 16 moves toward the joint 12 of the toggle lever 8 so that the lever is deflected in the same direction. The deflection is of sufficient extent such that the self-locking of the toggle lever system 8, 9 is overcome to permit the spring 4 to accelerate the system S to the left as viewed in FIG. 2. This movement continues until the right-hand toggle lever 9 stops the system S in the left-hand position as a result of the joint 13 abutting the abutment 15. In this position, the toggle lever 9 is located in an extended self-locking position. In addition, in the examples selected, the rotary slide valve (not shown) has reached a second discrete end position. A return movement to the initial position of FIG. 2 is triggered by the control device 20 sending a signal to the right-hand abutment 14 with the return movement of oscillation proceeding in the manner hereinbefore described.

The energy delivered by the abutments (i.e. solenoids) 14, 15 is greater than the energy dissipated in the complete system S so that there is no overall shortfall of energy. The required energy is supplied by the current injected into the winding 19.

The toggle levers 8, 9 can amplify the forces provided by the drive depending upon the coefficient of friction within the complete system, from one hundred-fold to one thousand-fold.

Referring to FIG. 3, wherein like reference characters indicate like parts as above, the drive may be used for an oscillatory mechanical system which is in the form of a linearly reciprocated system, for example, for a sliding spool valve. In this case, the oscillatory mechanical system includes a casing 31 including an inlet bore 34, a control bore 35 and a discharge bore 36 as well as a spool 25 which is slidably mounted in the casing 31 to selectively communicate the control bore 35 with one or the other of the inlet bore 34 and outlet bore 36. In addition, a prestressing means in the form of a pair of spring strips 26, 27 is provided for biasing the spool 25 from one end position to another end position. As shown, these spring strips 26, 27 are rigidly secured to a cross-member 30 in cantilever manner and abut against opposite ends of the spool 25.

The spool 25 may otherwise be part of a hydraulic system so as to control the flow of a fluid to and from a chamber or the like connected with the control bore 35 in the casing 31.

The drive for the sliding spool valve includes a pair of toggle levers 28, 32; 29, 33 each of which is pivotally secured to an opposite end of the spool 25 and pivotally secured to fixed points 10, 11 at opposite ends.

The operation of the drive is similar to that described above with respect to FIGS. 1 and 2. That is, with the spool 25 in the end or operative position illustrated in FIG. 3, the control bore 35 communicates via a groove 37 in the spool with the discharge bore 36. In this position, the spring 27 is prestressed while the toggle lever 28, 32 is in an extended self-locking position abutting against the armature of the left-hand abutment 14. When a control signal is delivered via the signal line 21 to the abutment 14, the toggle lever 28, 32 is deflected. This permits the spring 27 to slide the spool 25 within the casing 31 to the left, as viewed. At the same time, the toggle lever 29, 33 approaches the extended position against the abutment 15 while the spring 26 is flexed into a stressed condition. Upon reaching the second operative position, the spool 25 permits communication between the control bore 35 and the inlet bore 34.

The invention thus provides a drive for an oscillatory mechanical system having two discrete operative positions wherein the system experiences resilient prestressing in both of the operative positions and which is

moved during an adjustment or changeover with reduced friction while remaining substantially free from external disturbing forces. In this regard, the term "external disturbing forces" is intended to mean, for instance, retarding or driving forces caused by gases or liquids.

The invention further provides a drive of relatively simple construction which uses minimal energy to effect actuation of the system.

What is claimed is:

1. In combination

an oscillatory mechanical system having a first means for oscillating between two discrete operative positions and a resilient prestressing means for biasing said first means from one position to the other position; and

a drive for actuating said system, said drive including a pair of toggle levers connected to said first means, each said toggle lever being disposed to locate said first means in a respective operative position corresponding to an extended self-locking position of said respective toggle lever, a pair of movable abutments, each said abutment being disposed in the path of a respective toggle lever to inhibit overshooting of said respective toggle lever past said extended position thereof, and a control mechanism for selectively moving said abutments to effect a deflection of a respective toggle lever from an extended self-locking position thereof.

2. The combination as set forth in claim 1 wherein said first means includes a rotary spindle and a two-

armed lever secured to said spindle, said lever having one arm thereof pivotally connected to said pair of toggle levers.

3. The combination as set forth in claim 2 wherein said prestressing means includes a flexible spring secured at one end to a second arm of said two-armed lever and a resiliently deformable mount secured to said spring at an opposite end thereof.

4. The combination as set forth in claim 3 wherein each abutment includes an electromagnet having a movable armature for abutting a respective toggle lever.

5. The combination as set forth in claim 1 wherein said first means includes a sliding spool valve having a casing including an inlet bore, a control bore and a discharge bore and a spool slidably mounted in said casing to selectively communicate said control bore with one of said inlet bore and said outlet bore, said spool being connected at opposite ends to a respective toggle lever.

6. The combination as set forth in claim 5 wherein said prestressing means includes a pair of spring strips, each said spring strip being disposed at a respective end of said spool to bias said spool into said casing.

7. The combination as set forth in claim 6 wherein each abutment includes an electromagnet having a movable armature for abutting a respective toggle lever.

8. The combination as set forth in claim 1 wherein said first means is linearly reciprocal.

* * * * *

United States Patent [19]

Sakakibara et al.

[11] **4,446,889**

[45] **May 8, 1984**

[54] **PRESSURE MODULATING VALVE ASSEMBLY**

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[73] Assignee: **Aisin Seiki Kabushiki Kaisha, Kariya, Japan**

[21] Appl. No.: **404,738**

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[30] **Foreign Application Priority Data**

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[51] **Int. Cl.**^{3} **F15B 13/044; F16K 11/00; F16K 11/02**

[52] **U.S. Cl.** **137/625.4; 251/138; 335/270; 335/274**

[58] **Field of Search** 251/138, 141; 137/625.4, 625.65; 335/135, 2, 270, 274, 275, 276

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4,249,457	2/1981	Sakakibara	137/625.65 X
4,250,924	2/1981	Sakakibara	251/138 X

Primary Examiner—Martin P. Schwadron
Assistant Examiner—John S. Starsiak, Jr.
Attorney, Agent, or Firm—Sughrue, Mion, Zinn, Macpeak and Seas

[57] **ABSTRACT**

The pressure modulating valve assembly includes a solenoid coil, a yoke having curved support surfaces extending from one end of the coil to the other, and a resilient non-magnetic valve support having at least one valve secured thereto and a pair of attaching legs secured to the yoke in overlying relation relative to the curved support surfaces. An armature of magnetic material is secured to the valve support at a central portion thereof spaced from the point of attachment of the valve support to the yoke. A spring normally biases the armature in a direction away from the solenoid coil so that upon energization of the coil the solenoid support and the armature secured thereto are subjected to a rolling motion along the curved surfaces of the yoke to provide a friction free connection between the yoke and the combined armature and valve support.

5 Claims, 4 Drawing Figures

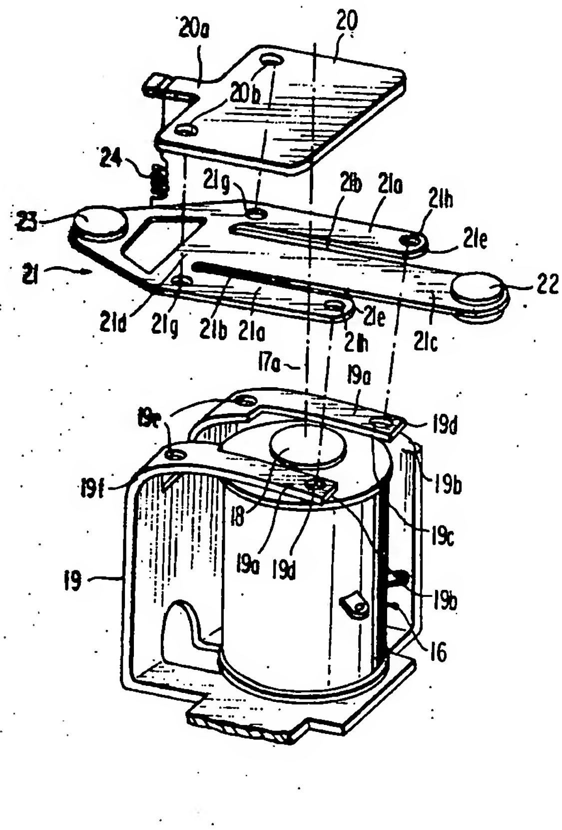

FIG. 1

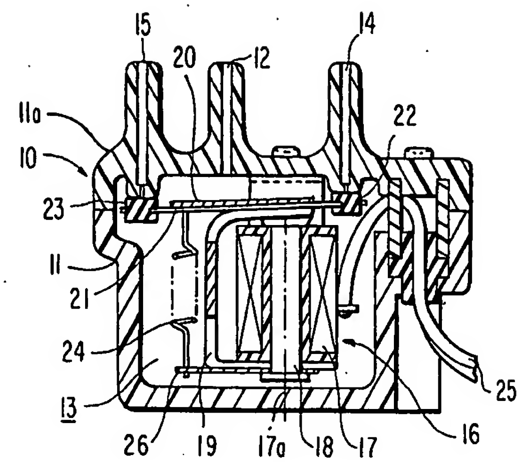
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FIG. 2

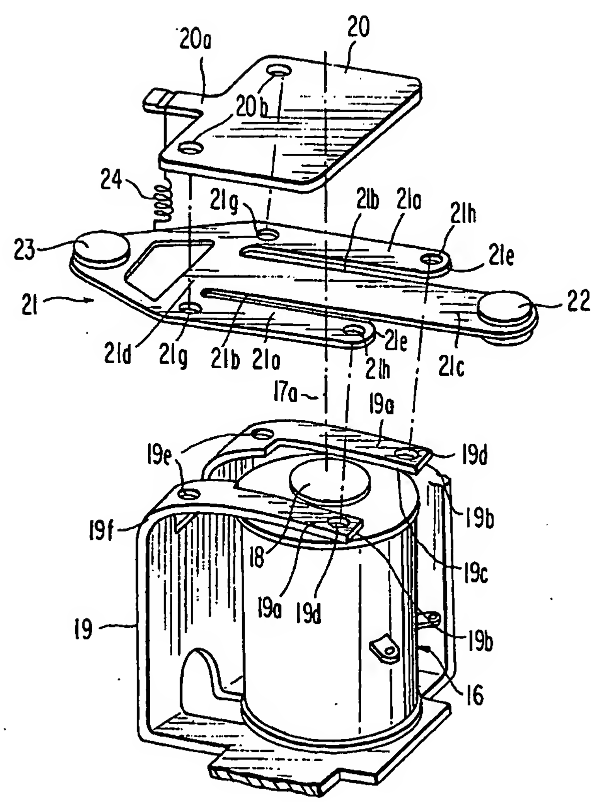
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FIG. 3

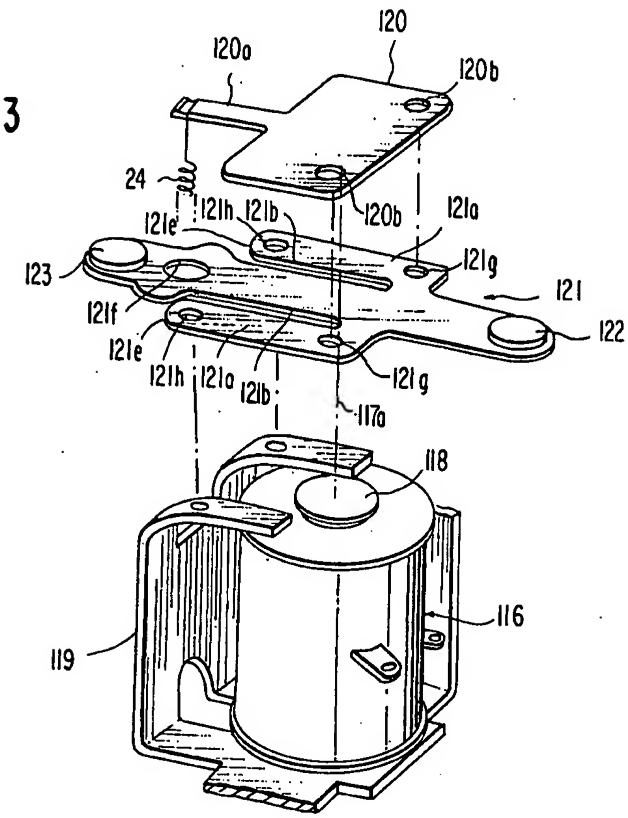
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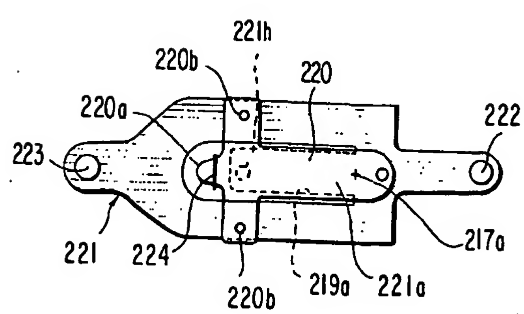
FIG. 4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PRESSURE MODULATING VALVE ASSEMBLY

BACKGROUND OF THE INVENTION

The present invention is directed to a pressure modulating valve assembly and more particularly to an electrically operated valve supporting member which is mounted in rolling contact with a yoke of a solenoid coil in response to the operation of the solenoid coil to modulate pressure in a valve chamber.

A conventional pressure modulating valve is disclosed in U.S. Pat. No. 4,249,457 granted Feb. 10, 1981 and entitled "Vacuum Servo Motor". In the conventional pressure modulating valve assembly, an armature of magnetic material having recesses at opposite sides thereof is pivoted on a yoke member by means of upstanding projections formed at opposite sides of the yoke whereby the projections are loosely fitted within the recesses, respectively. A valve supporting member of non-magnetic material is secured to the armature and is provided with valve members at opposite ends thereof. Spring means are provided for biasing the armature to a first position wherein one of the valve members engages a valve seat and a solenoid coil is provided for biasing the armature to a second position against the force of the spring means to move the other valve member into engagement with a valve seat while moving the first mentioned valve member out of engagement with its respective valve seat. In the foregoing arrangement, the durability of the valve is lessened since the recesses of the armature and the projections on the yoke are quickly worn by the constant rubbing action which occurs on every rotation of the armature. Such wear eventually leads to undesirable tolerances which adversely affect the seating of the valve members on their respective seats.

SUMMARY OF THE INVENTION

The present invention provides a new and improved pressure modulating valve assembly which obviates the above-mentioned disadvantages associated with the conventional pressure modulating valve assembly.

The present invention provides a new and improved pressure modulating valve assembly which is low in cost, simple in construction and durable in operation.

The present invention provides a new and improved pressure modulating valve assembly comprising a solenoid coil, yoke means having curved support surfaces extending from one end of said coil to the other, resilient nonmagnetic valve supporting means having at least one valve secured thereto and attaching leg means secured to said yoke means in overlying relation relative to said curved supporting surfaces, an armature of magnetic material secured to said valve supporting means at a central portion thereof spaced from the point of attachment of said valve supporting means to said yoke means and spring means normally biasing said armature in a direction away from said solenoid coil whereby upon energization of said coil said solenoid supporting means and said armature secured thereto are subjected to a rolling motion along the curved surface of said yoke means to provide a friction-free connection between said yoke means and said combined armature and valve support means.

The foregoing and other objects, features and advantages of the invention will be apparent from the following more particular description of a preferred embodiment f the invention as illustrated in the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross-sectional view through a pressure modulating valve assembly according to the present invention showing a first embodiment of a valve supporting and operating arrangement.

FIG. 2 is an exploded perspective view showing the principal parts of the invention according to the first embodiment of FIG. 1.

FIG. 3 is an exploded perspective view similar to FIG. 2 showing a second embodiment of the present invention.

FIG. 4 is a plan view showing a valve supporting plate and armature assembly according to a third embodiment of the present invention.

DETAILED DESCRIPTION OF THE INVENTION

The pressure modulating valve assembly **10** as shown in the embodiment of FIG. 1, is provided with a casing **11** and a cover **11*a*** having a pair of inlet ports **14, 15** and an outlet port **12**. The cover **11*a*** is adapted to be hermetically engaged with the casing **11** to define a valve chamber **13** therein. An electromagnet assembly **16** comprised of a core **18** and a solenoid coil **17** is carried by a yoke **19** which in turn is secured to the cover **11*a***. The top portion of the yoke **19** is provided with a pair of curved arms **19*a*, 19*a*** having free end portions **19*b*, 19*b*** extending over the solenoid coil **17** on opposite sides of the central axis **17*a***. The lower end of the core **18** is secured to the bottom portion of the yoke **19** and the lead wires **25** connected to the coil **17** are hermetically sealed where they pass through an opening in the casing **11**. An elongated flexible, resilient valve supporting member **21** of non-magnetic material is adapted to be secured to the curved arms **19*a*, 19*a*** as best seen in the exploded view of FIG. 2. A pair of valve members **22** and **23** are secured to opposite ends of the valve supporting member at positions corresponding to the positions of the inlet ports **14** and **15**, respectively. The valve supporting member **21** is provided with a pair of elongated legs **21*a*, 21*a*** which extend parallel to the main body **21*c*** of the valve supporting member on opposite sides thereof. The leg portions **21*a*, 21*a*** are spaced from the body portion **21*c*** by slots **21*b*, 21*b***. The leg portions **21*a*, 21*a*** are provided with free end portions **21*e*, 21*e*** and are of integral onepiece construction with a transverse connecting portion **21*d*** at the opposite ends thereof. An opening **21*f*** is provided intermediate the valve member **23** and the transversely extending connecting portion **21*d*** to provide a passage for a spring **24**, the arrangement and purpose of which will be described hereinafter.

A substantially rectangular armature **20** of magnetic material is secured to the valve supporting member by means of pins or rivets (not shown) extending through a pair of holes **21*g*, 21*g*** located in the leg portions **21*a*, 21*a*** adjacent the transversely extending connecting portion **21*d*** and a pair of holes **20*b*, 20*b*** formed in the armature **20**. The valve supporting member **21** is in turn secured to the yoke **19** by pins or rivets (not shown) which extend through a pair of holes **21*h*, 21*h*** formed in the free ends **21*e*, 21*e*** of the leg portions **21*a* 21*a*** and a pair of holes **19*d*, 19*d*** located adjacent the free ends **19*b*, 19*b*** of the curved arms **19*a*, 19*a*** of the yoke **19**. An additional pair of holes **19*e*, 19*e*** are provided in the curved

portions of the arms **19a**, **19a** f r receiving the ends of the pins or rivets which secure the armature to the valve supporting member. With the armature **20**, the valve supporting member **21** and the yoke **19** secured in the f regoing manner, the armature **20** is positioned in overlying relation relative to the curved arms of the yoke **19** with the portion of the armature **20** closest to the valve member **22** being disposed in overlying relation relative to the core **18** of the electromagnet **16**. A projection **20a** extends outwardly from the opposite side of the armature **20** and one end of the spring **24** is secured to the projection **20a**. The opposite end of the spring **24** is secured to a retainer plate **26** which is secured between the core **18** and the yoke **19** of the electromagnet assembly **16**. The biasing force of the spring **24** urges the projection **20a** of the armature **20** downwardly thereby maintaining the desired gap between the body **21c** of the valve supporting member **21** and the core **18** when the solenoid coil **17** is in the deenergized condition. In this position, the valve member **22** is disposed in engagement with the valve seat surrounding the inlet port **14** and the valve member **23** is in spaced relation to the valve seat surrounding the inlet port **15**. The central axis **17a** of the electromagnet assembly **16** is positioned between the arms **19a**, **19a** of the yoke **19** whereby upon energization of the solenoid coil the armature **20** will be moved toward the electromagnet **16** against the biasing force of the spring **24**. As the armature **20** and the valve support plate **21** which are secured together are drawn toward the electromagnet **16** the valve supporting plate pivots about a line extending transversely across the arms **19a**, **19a**, which line moves progressively along the curved surface of the arms. Thus, the valve supporting plate **21** and the armature **20** effectively rock on the curved arms with a rolling motion which is substantially friction free thereby eliminating any possible wear between the arms of the yoke **19** and the valve supporting plate **21**. The pivotal or rolling motion of the armature **20** continues until such time as the valve member **23** engages the seat surrounding the inlet port **15** and the valve member **22** is disengaged from the seat surrounding the inlet port **14**.

When the solenoid coil **17** is deenergized the valve supporting plate **21** and the armature **20** pivot in a rolling fashion about the curved arms of the yoke **19** under the influence of the spring **24** to return the valve members to the original condition. In the valve assembly according to the present invention the inlet port **14** may be in communication with a vacuum source such as the intake manifold of an automobile and the inlet port **15** may be in communication with the atmosphere. The valve members **22** and **23** are alternately moved into seating engagement with the valve seats surrounding the inlet ports **14** and **15** in accordance with the energization and deenergization of the solenoid coil which receives an electrical input signal such as a pulse frequency signal from a suitable source. Thus, the pressure in the valve chamber is modulated in accordance with the operation of the valve members **22** and **23**. The modulated pressure in the valve chamber **13** is applied to a suitable actuator (not shown) through the outlet port **12**.

According to a second embodiment of the present invention as shown in FIG. 3, the valve supporting member **121** is so constructed that the transversely extending connecting portion **121d** thereof is arranged closer to the valve member **122** which is substantially opposite to the arrangement shown in the embodiment of FIGS. **1** and **2**. The armature **120** is connected to the transversely extending connection portion **121d** by suitable rivets or the like (not shown) in a manner similar to that disclosed in the previous embodiment. The various other components f the valve assembly according to the embodiment of FIG. **3** which are identified by reference numerals in the 100 series correspond to similar elements in the embodiment of FIG. **2** with corresponding lower order reference numerals. The operation of the assembly according to the embodiment of FIG. **3** is substantially identical to that described above with respect to the embodiment of FIG. **2**.

In the third embodiment of the present invention as shown in FIG. **4**, the valve supporting member **221** is provided with valve members **222** and **223** at opposite ends thereof and an elongated central recess in which a single leg portion **221a** extends along the axis between the two valve members. The free end of the leg portion **221a** is provided with an aperture **221h** through which a rivet (not shown) or the like extends for securing the free end of the leg portion **221a** to the yoke having a single arm **219a** adapted to overlie the core of an electromagnet whose axis would be located at the point **217a**. An armature **220** is secured to the valve supporting member **221** by means of rivets **220b**, **220b** or the like and is provided with a projection **220a** to which one end of the spring **224** is secured. The operation of the valve members on the valve supporting member is substantially identical to that described with respect to the previous embodiments

In summary, the pressure modulating valve assembly **10** is not subjected to any frictional rubbing action between moving members since the armature is connected to the valve supporting member and the yoke in a manner whereby a friction-free rolling action is obtained. The pressure modulating valve assembly according to the present invention is capable of modulating not only vacuum pressure but also positive pressure. Liquid pressure may also be modulated by a valve assembly according to the present invention.

While the invention has been particularly shown and described with respect to preferred embodiments thereof, it will be understood by those in the art that the foregoing and other changes in form and details may be made therein without departing from the spirit and scope of the invention.

What is claimed is:

1. A pressure modulating valve assembly comprising housing means having at least one inlet port and one outlet port, solenoid coil means mounted in said housing means, magnetic yoke means associated with said coil means and having curved arm means extending over one end of said coil means, a resilient non-magnetic valve supporting member having leg means secured to said yoke means in overlying relation to said curved arm means and said solenoid coil means, armature means secured to said valve supporting means at a location spaced from the securement of said valve supporting member to said yoke means with said armature overlying said solenoid coil means, spring means connected to said armature means for normally biasing said armature means and valve supporting member about said curved surface of said yoke means away from said solenoid coil means and at least one valve member on said valve supporting member adapted to be moved into and out of engagement with valve seat means surrounding said inlet port.

2. A pressure modulating valve assembly as set forth in claim 1, further comprising a second valve member carried by said valve supporting member, said valve members being located at opposite ends of said valve supporting member on opposite sides of said armature means.

3. A pressure modulating valve assembly as set forth in claim 1, wherein said magnetic yoke means has a substantially C-shaped configuration with one end thereof secured to one end of said solenoid coil and the opposite end thereof being in the form of a pair of parallel curved arms extending over the opposite end of said solenoid coil means and wherein said valve supporting member is comprised of a flat resilient non-magnetic elongated body having a transversely extending cross piece adjacent one end thereof with a pair of flat, resilient elongated legs extending therefrom toward the other end of said body in parallel spaced relation thereto on opposite sides thereof, the free ends of said legs being secured to the ends of said curved arms of said yoke means.

4. A pressure modulating valve assembly as set forth in claim 3, wherein said armature is secured to the opposite surfaces of said legs adjacent said cross piece with the armature overlying said body and said legs of said valve support member.

5. A pressure modulating valve assembly as set forth in claim 1, wherein said yoke means is comprised of a C-shaped member secured at one end thereof to one end of said solenoid coil means and with the opposite end comprising a single curved leg extending over the opposite end of said solenoid coil means and wherein said valve supporting member is comprised of an elongated body member having an elongated opening in the central portion thereof with a single leg integral with said body extending in said opening substantially the entire length thereof, said leg being secured at its free end to said curved arm of said yoke means and said armature being secured to the opposite surface of said valve supporting member in overlying relation to said opening.

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United States Patent [19]

Chikamatsu et al.

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[45] **Date of Patent: May 10, 1994**

[54] **ELECTROMAGNETIC PROPORTIONAL PRESSURE REDUCING VALVE**

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[73] Assignees: **Kayaba Kogyo Kabushiki Kaisha; Nissan Motor Co., Ltd.,** both of Yokohama, Japan

[21] Appl. No.: **64,198**

[22] Filed: **May 19, 1993**

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[52] U.S. Cl. **137/625.65**; 91/433; 251/69; 251/129.08; 251/129.1

[58] Field of Search 91/433; 137/625.65; 251/69, 129.08, 129.1

[56] **References Cited**

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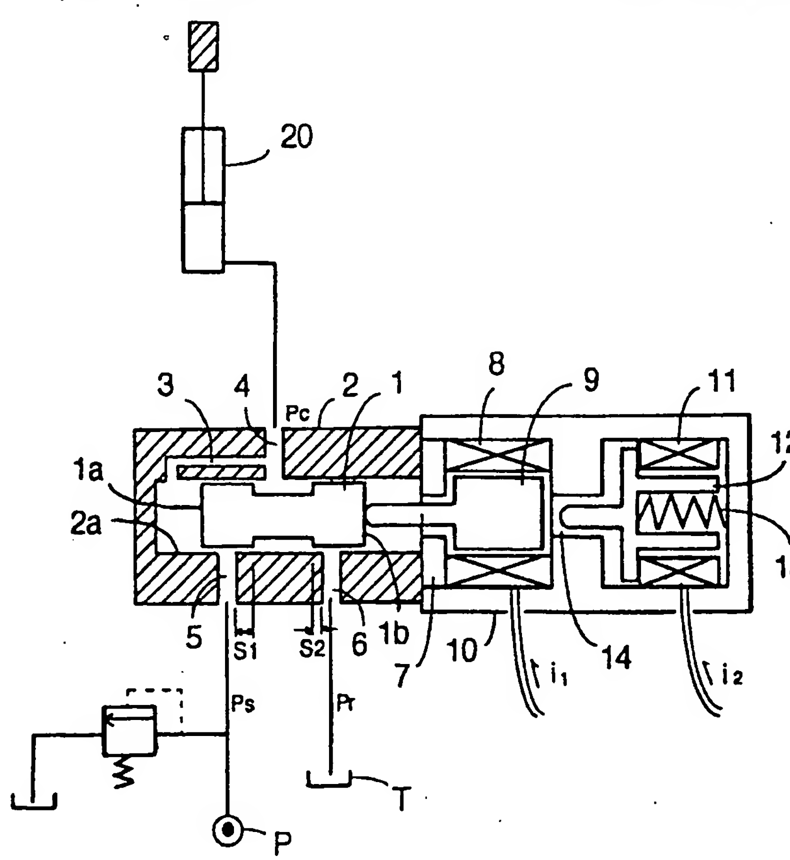
Primary Examiner—Gerald A. Michalsky

Attorney, Agent, or Firm—Foley & Lardner

[57] **ABSTRACT**

This invention relates to a pressure reducing valve wherein a spool is driven axially by a first plunger which moves according to the magnetization state of a first solenoid, and a control port is connected to a supply port and a tank port according to the axial position of this spool. The pressure of the control port acts on the spool in the opposite direction to that of the plunger via a feedback passage, and the pressure increases or decreases according to the current flowing through the first solenoid. According to this invention, a second plunger is arranged in series with the first plunger, and a spring tends to push the second plunger toward the first plunger. A second magnetized solenoid however keeps the second plunger held back against the restoring force of the spring so that it is not in contact with the first plunger, and releases the second plunger so that it pushes the first plunger when the current fails. Hence, even if the current supplied to the first and second solenoids is cut off, the control port maintains the predetermined pressure, and a fail-safe mechanism is realized by means of a simple construction.

3 Claims, 5 Drawing Sheets



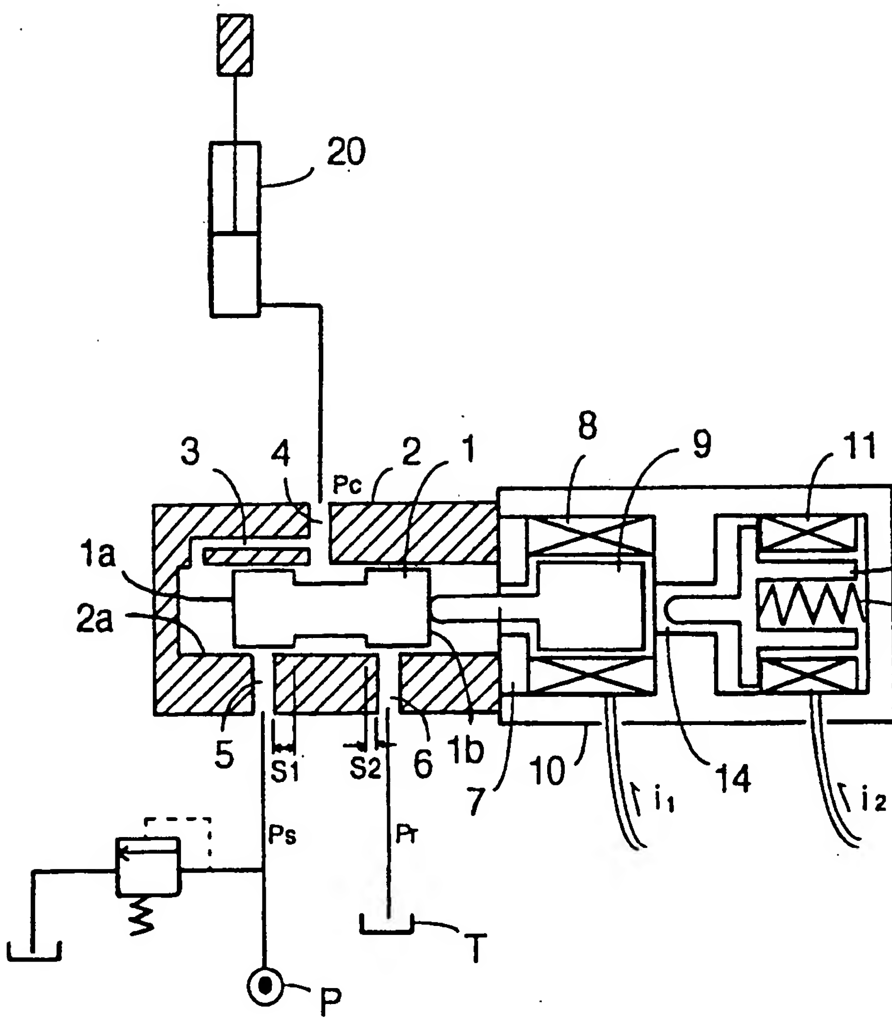


FIG. 1

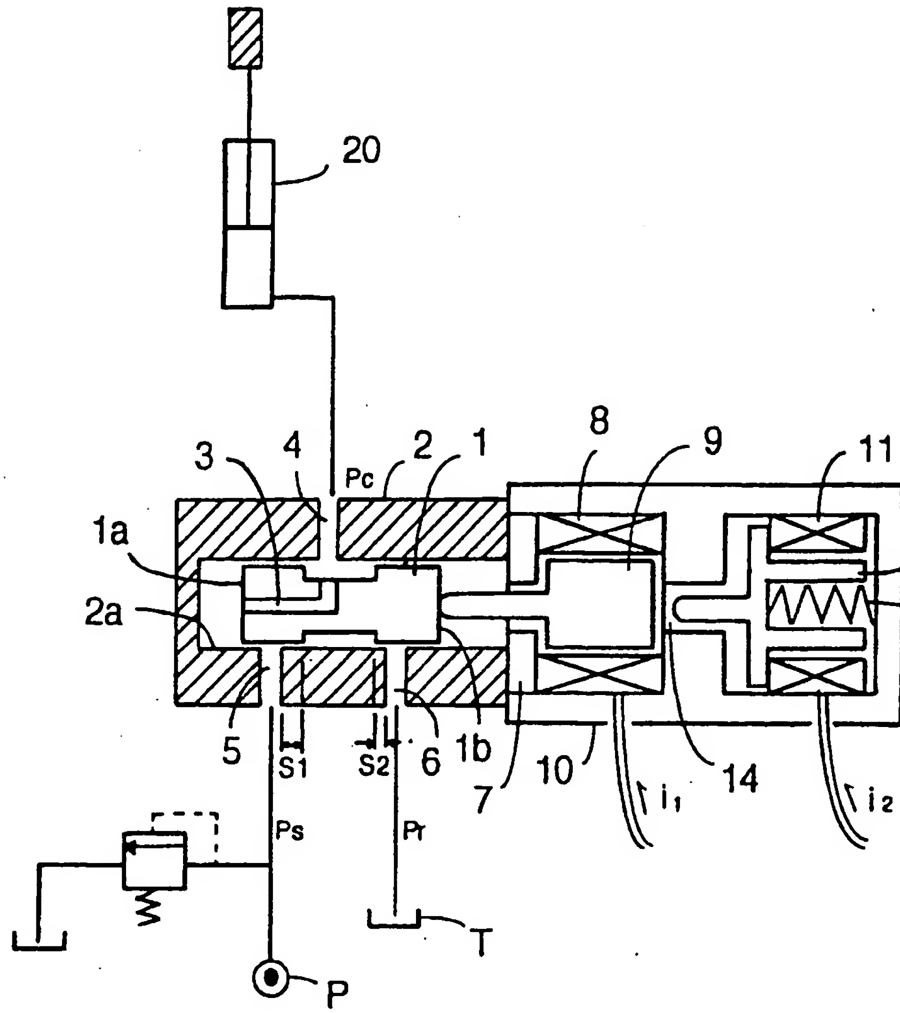


FIG. 2

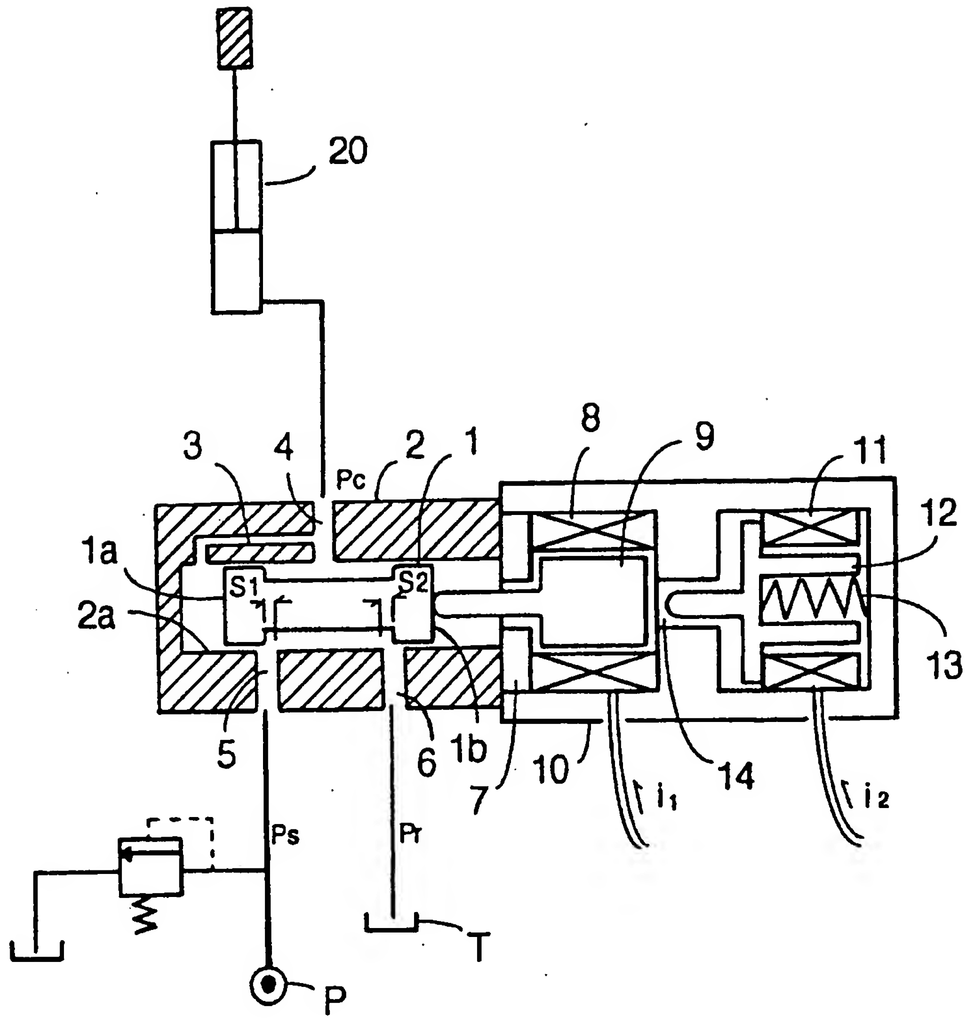


FIG. 3

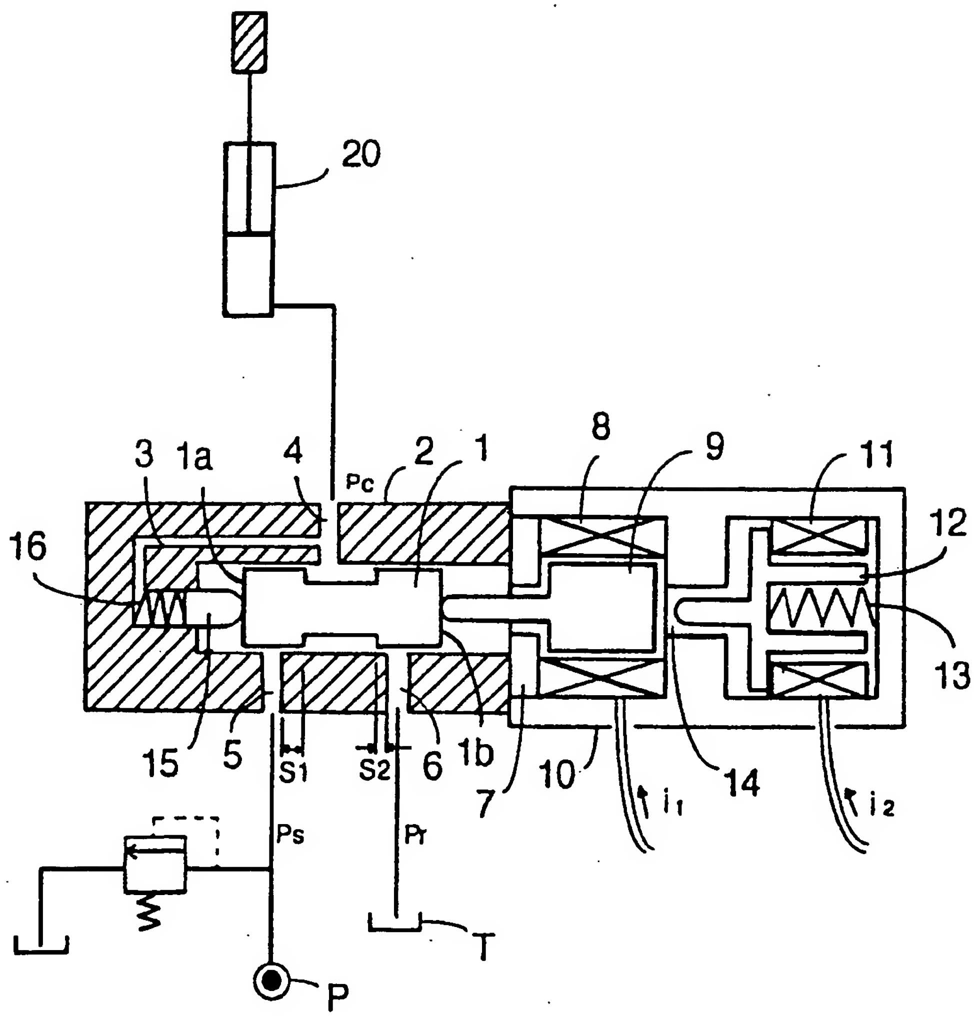
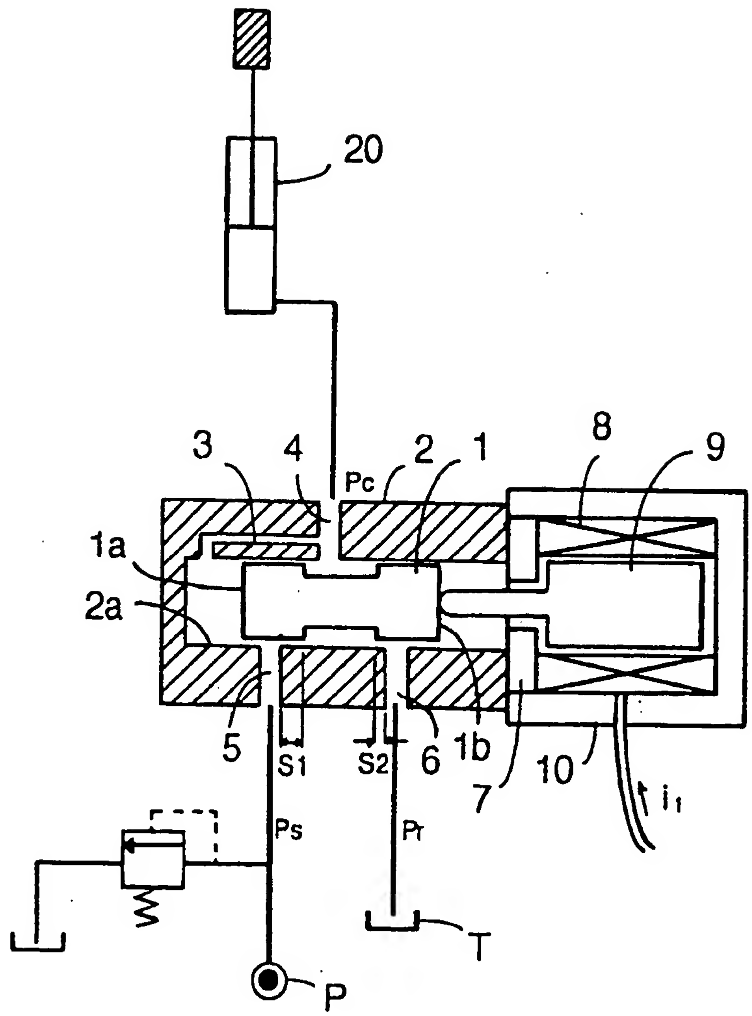


FIG. 4



PRIOR ART

FIG. 5

ELECTROMAGNETIC PROPORTIONAL PRESSURE REDUCING VALVE

FIELD OF THE INVENTION

This invention relates to the fail-safe function of an electromagnetic proportional pressure reducing valve.

BACKGROUND OF THE INVENTION

An electromagnetic pressure reducing valve, as shown in FIG. 5 for example, comprises a solenoid **8** which causes a spool **1** to slide for connecting a control port **4** to either a supply port **5** or a tank port **6** depending on the slide position.

The spool **1** is inserted such that it is free to slide in a hole *2a* in a valve housing **2**. The control port **4**, supply port **5** connected to an oil source P and tank port **6** connected to a tank T open into this cavity, and a feedback oil passage **3** which branches off the control port **4** enables a pressure Pc of the control port **4** to be applied to one end *1a* of the spool **1**.

The solenoid **8** is housed in a case **10** connected to the valve housing **2**, and a plunger **9** is made to support the opposite end *1b* of the spool **1** according to the magnetization state of the solenoid **1**. When an electric current is passed through the solenoid **8**, the solenoid **8** is magnetized, and the plunger **9** is thereby pulled toward a base **7** of the case **10** so as to push the spool **1** towards the left of the figure. The pushing force in this case is proportional to the magnetizing current flowing through the solenoid **8**.

Small gaps exist between the hole *2a* and the spool **1** so that the control port **4**, supply port **5** and tank port **6** are interconnected. The lengths S_1 and S_2 of these gaps in the axial direction vary according to the position of the spool **1**, however S_1+S_2 is always constant.

The pressure Pc generated in the control port **4** varies according to the ratio of S_1 and S_2. For example, when S_1 is small and S_2 is large, the pressure Pc increases.

If a current i_1 is supplied to the solenoid **8**, a driving force F_1 acts on the spool **1** tending to push it toward the left of the figure. Further, as the pressure Pc acts on the left end of the spool **1** via the feedback oil passage **3**, a pushing force F_2 based on this pressure Pc acts on the spool **1** in a right hand direction. If the pressure receiving surface area of the spool **1** is Aa, this pushing force is given by:

$$F_2 = Pc \times Aa \quad (1)$$

The spool **1** is thus held in a position at which the driving force F_1 due to the solenoid **8** and the pushing force F_2 acting in the reverse direction are balanced. The driving force F_1 is proportional to the electric current i_1 flowing through the solenoid **8**, while the pushing force F_2 is proportional to the pressure Pc generated by the control port **4**. In the balanced state:

$$Pc = F_1/Aa \quad (2)$$

The pressure Pc is therefore proportional to the current i_1 flowing through the solenoid **8**. In other words, if the magnetizing current flowing through the solenoid **8** increases or decreases, the spool driving force F_1 of the plunger **9** varies, so the spool is displaced to a position at which the pushing force F_2 is in equilibrium with it, and the ratio of S_1 to S_2 varies. By varying the ratio of S_1 to S_2 in this manner, a high pressure supplied to the

supply port **5** is reduced to an arbitrary control pressure Pc.

However, if in such an electromagnetic proportional reducing valve the current flowing through the solenoid **8** is interrupted for some reason, the spool driving force F_1 of the plunger **9** drops to zero, the spool moves to the right of the figure as far as it can, and the control pressure Pc falls to the pressure Pr of the tank port **6**.

If the magnetizing current i_1 is interrupted due to a fault when a hydraulic actuator **20** is supporting a load due to the pressure Pc, the control port **4** is no longer to maintain the pressure, and the hydraulic actuator **20** shortens. If the actuator **20** is used in a hydraulic unit for distributing the driving force in a torque split type four-wheeled vehicle and power is no longer supplied to the solenoid **8** due to a fault in the electric circuit, therefore, four-wheeled drive becomes impossible.

To prevent this, another pressure valve could be provided for the event of an oil pressure emergency situation, but this would make the hydraulic circuit more complex and inevitably lead to greater cost.

Alternatively, the plunger **9** could be pulled toward the spool **1** by a spring so that even if the spool driving force F_1 dropped to zero when the solenoid magnetizing current i_1 was interrupted, the control port **4** maintained a certain minimum pressure, but in this case it would no longer be possible by ordinary means to set the pressure Pc of the control port **4** below this minimum level.

SUMMARY OF THE INVENTION

It is therefore an object of this invention to maintain a predetermined control pressure when the solenoid magnetizing current in an electromagnetic proportional pressure reducing valve is interrupted due to a fault or to other circumstances.

It is a further object of this invention to maintain the aforesaid control pressure when the magnetizing current is accidentally interrupted.

It is a further object of this invention to implement the aforesaid fail-safe mechanism by means of a simple and economical construction.

It is yet another object of this invention to maintain the minimum pressure of a torque split actuator fitted to a four wheel drive vehicle when the electrical system is faulty, and thereby, to maintain the four wheel driving state of the vehicle.

In order to achieve the above object, this invention provides a pressure reducing valve comprising a supply port connected to an oil source, a tank port connected to a tank, a control port connected to a hydraulic actuator, a spool connecting the control port to the tank port depending on its axial position, a first plunger in contact with one end of the spool, a first solenoid which drives the first plunger axially by means of a driving force according to an magnetizing current provided, and a feedback oil passage for applying the pressure of the control port to the opposite end of the spool, wherein the pressure generated by the control port is increased and decreased according to the magnetizing current of the first solenoid. This valve further comprises a second plunger arranged in series with the first plunger, a spring which pushes the second plunger in the direction of the first plunger, and a second solenoid for pulling the second plunger in opposition to the restoring force of the spring such that it is not in contact with the first plunger when the second solenoid is magnetized.

It is preferable that the feedback passage is formed in the spool.

It is also preferable that the valve further comprises a pin having a pressure receiving surface area smaller than that of the spool which receives pressure from the feedback passage so as to support the opposite end of the spool.

The details as well as other features and advantages of this invention are set forth in the remainder of the specification and are shown in the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a vertical sectional view of an electromagnetic proportional reducing valve according to a first embodiment of this invention.

FIG. 2 is a vertical sectional view of an electromagnetic proportional reducing valve according to a second embodiment of this invention.

FIG. 3 is a vertical sectional view of an electromagnetic proportional reducing valve according to a third embodiment of this invention.

FIG. 4 is a vertical sectional view of an electromagnetic proportional reducing valve according to a fourth embodiment of this invention.

FIG. 5 is a vertical sectional view of an electromagnetic proportional reducing valve according to the prior art.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring to FIG. 1 of the drawings, a spool 1 is inserted in a hole 2*a* inside a valve housing 2 such that it can freely slide in the housing. A control port 4, supply port 5 connected to an oil source P and tank port 6 connected to a tank T open into this hole 2*a*, and the pressure Pc generated by the control port 4 is controlled according to the position of the spool 1.

A feedback oil passage 3 which branches off the control port 4 exerts the pressure Pc of the control port 4 on one end 1*a* of the spool 1.

A plunger 9 driven by a first solenoid 8 and a plunger 12 driven by a second solenoid 11, the second plunger being arranged coaxially and in series with the first plunger, are disposed in a case 10 connected to the value housing 2. A throughhole 14 facing the rear of the plunger 9 is formed in the case 10, the tip of the second plunger 12 being inserted in this throughhole 14.

When the solenoid 11 is magnetized, the second plunger 12 is held back against the force of a spring 13 such that it is not in contact with the first plunger 9.

However, when power to the solenoid 11 is interrupted, it moves forward due to the restoring force of the spring 13 so as to push the plunger 9 from the rear and thereby push the spool 1 to a predetermined position.

A control current i_1 which varies according to a target pressure value of the control port 4 is supplied to the first solenoid 8, and a constant current i_2, which is larger than the minimum current required to maintain the second plunger 12 held back against the force of the spring 13, is supplied to the second solenoid 11.

In the normal state, the second solenoid 11 which is magnetized by the current i_2 pulls the plunger 12 so that it is not in contact with the plunger 9.

The control current i_1 which varies according to the target pressure is applied to the first solenoid 8, and the plunger 9 displaces the spool 1 due to the driving force produced. A pressure dependent on the position of the spool 1 is thereby generated by the control port 4, and normal pressure reduction control is performed.

If the power to the first and second solenoids 8, 11 is interrupted due to an electrical fault or to any other reason, the spool driving force of the plunger 9 falls to zero, and the spool 1 is pushed back by the feedback pressure.

However, as the plunger 12 is no longer restrained by the solenoid 11, this plunger 12 moves forward due to the restoring force of the spring 13. The first plunger 9 is therefore returned by a predetermined force, and the spool 1 comes to rest at a position determined by the set force of the spring 13. As a result, the control pressure in the control port 4 does not fall to the tank port pressure Pr, and the pressure is maintained according to the position of the spool 1.

Even if the current is interrupted, the control port 4 maintains a constant pressure, and a hydraulic actuator 20 can continue to support a load by means of this pressure.

Further, even if the current is deliberately shut off when, for example, a device comprising the hydraulic actuator 20 also comprises other components and a fault is detected in these other components, the pressure of the hydraulic actuator 20 can be maintained.

On the other hand, if normal control is terminated, the pressure Pc of the control port 4 can be reduced to the pressure of the tank port 6. In other words, if power to the first and second solenoids is interrupted and the supply of actuating oil from the oil source P is simultaneously stopped, oil pressure does not appear at the supply port 5 even if the spool 1 is maintained at its predetermined position. The pressure Pc of the control port 4 is then equivalent to the pressure of the tank port 6.

If for example this hydraulic actuator 20 is used in a hydraulic unit which distributes the driving force in a torque split type vehicle having four wheel drive, and a fault occurs in the vehicle's electrical system, the minimum four wheel driving force can be maintained, and the driving force is distributed. This distribution would be in such a ratio that tight corner braking does not occur.

When the power supply to the first and second solenoids 8, 11 is resumed, the plunger 12 withdraws to a position at which it no longer interferes with the first plunger 9, the position of the spool 1 is determined only by the first plunger 9 which is displaced according to the control current i_1, and normal control is restored.

The pressure can be maintained simply by providing a second solenoid 11 as described hereintofore. This arrangement therefore constitutes a simple fail-safe mechanism without any need to provide a special pressure maintaining valve in the hydraulic circuit. Further, as this is achieved by providing a pressure reducing valve with a first and second solenoid in a coaxial arrangement, basic pressure reduction characteristics are exactly the same as in a pressure reduction valve according to the prior art, there is no risk that reliability of valve operation will be impaired, and the pressure reducing valve can be manufactured at low cost.

A second embodiment of this invention will now be described with reference to FIG. 2. According to this second embodiment, the feedback oil passage 3 which applies the pressure Pc generated by the control port 4 to the end 1*a* of the spool 1, passes right through the spool 1. There is thus no need to provide the feedback

oil passage 3 in the housing 2, and the structure of the pressure reducing valve can be made more compact.

FIG. 3 shows a third embodiment of this invention. This valve has an "underlap" type of spool of which the sliding surface does not completely cover the supply port 5 and the tank port 6, and in this case also, the pressure Pc is controlled according to the position of the spool 1.

FIG. 4 shows a fourth embodiment of this invention wherein a reaction pin 15 having a pressure receiving surface area As less than the pressure receiving surface area Aa of the spool 1, is provided to apply the pushing force F_2 to the end 1*a* of the spool 1 according to the pressure Pc of the control port 4.

The pressure Pc of the feedback oil circuit 3 acts on the reaction pin 15, which then pushes the spool 1. In this case, if the pressure receiving surface area of the pin 15 is As, this pushing force F_2 is given by:

$$F_2 = Pc \times As \qquad (3)$$

Since As<Aa, to oppose an identical driving force F_1, the generated control pressure Pc must be made larger. By using such a reaction pin 15, therefore, the driving force of the solenoid 8 required to generate the same pressure Pc is less, a smaller solenoid can be used, and the whole apparatus becomes more compact and economical. 16 is a return spring which maintains the reaction pin 15 in contact with the spool 1.

The foregoing description of the preferred embodiments for the purpose of illustrating this invention is not to be considered as limiting or restricting the invention, since many modifications may be made by those skilled in the art without departing from the scope of the invention.

The embodiments of this invention in which an exclusive property or privilege is claimed are defined as follows:

1. A pressure reducing valve comprising a supply port connected to an oil source, a tank port connected to a tank, a control port connected to a hydraulic actuator, a spool connecting said control port to said tank port depending on its axial position, a first plunger in contact with one end of said spool, a first solenoid which drives said first plunger axially by means of a driving force according to an magnetizing current provided, and a feedback oil passage for applying the pressure of said control port to the opposite end of said spool, wherein the pressure generated by said control port is increased and decreased according to the magnetizing current of said first solenoid, further comprising:

a second plunger arranged in series with said first plunger,

a spring which pushes said second plunger in the direction of said first plunger, and

a second solenoid for pulling said second plunger in opposition to the restoring force of said spring such that it is not in contact with said first plunger when said second solenoid is magnetized.

2. An electromagnetic proportional pressure reducing valve as defined in claim 1 wherein said feedback passage is formed in said spool.

3. A pressure reducing valve comprising a supply port connected to an oil source, a tank port connected to a tank, a control port connected to a hydraulic actuator, a spool connecting said control port to said tank port depending on its axial position, a first plunger in contact with one end of said spool, a first solenoid which drives said first plunger axially by means of a driving force according to a magnetizing current provided, and a feedback oil passage for applying the pressure of said control port to the opposite end of said spool, wherein the pressure generated by said control port is increased and decreased according to the magnetizing current of said first solenoid, further comprising:

a pin having a pressure receiving surface area smaller than that of said spool which receives pressure from said feedback passage so as to support said opposite end of said spool,

a second plunger arranged in series with said first plunger,

a spring which pushes said second plunger in the direction of said first plunger, and

a second solenoid for pulling said second plunger in opposition to the restoring force of said spring such that it is not in contact with said first plunger when said second solenoid is magnetized.

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US Full Image

US-PAT-NO: 0914094

DOCUMENT-IDENTIFIER: US 0914094 A

TITLE: OCR SCANNED DOCUMENT

DATE-ISSUED: March 2, 1909

INVENTOR: Name not available

US-CL-CURRENT: 251/129.09

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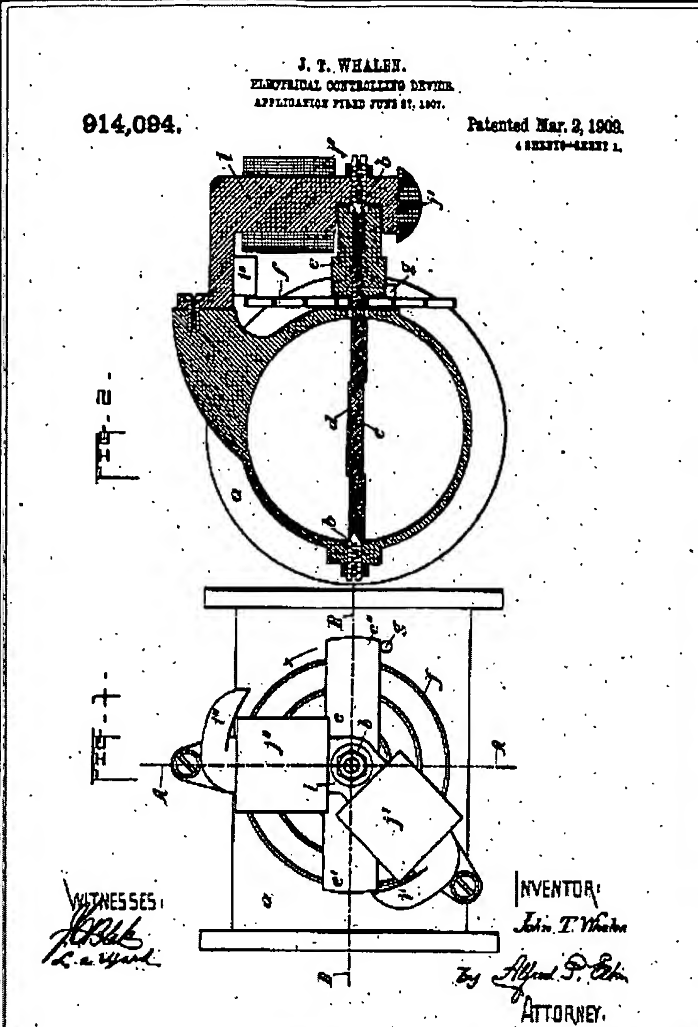

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US005868108A

United States Patent [19]

Schmitz et al.

[11] **Patent Number: 5,868,108**

[45] **Date of Patent: Feb. 9, 1999**

[54] **METHOD FOR CONTROLLING AN ELECTROMAGNETIC ACTUATOR OPERATING AN ENGINE VALVE**

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[73] Assignee: **FEV Motorentechnik GmbH & Co. KG,** Aachen, Germany

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[22] Filed: **Dec. 11, 1997**

[30] **Foreign Application Priority Data**

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Jun. 4, 1997 [DE] Germany 197 23 405.4

[51] **Int. Cl.**[6] **F01L 9/04**

[52] **U.S. Cl.** **123/90.11**; 251/129.01; 251/129.16

[58] **Field of Search** 123/90.11; 251/129.01, 251/129.02, 129.05, 129.1, 129.15, 129.16

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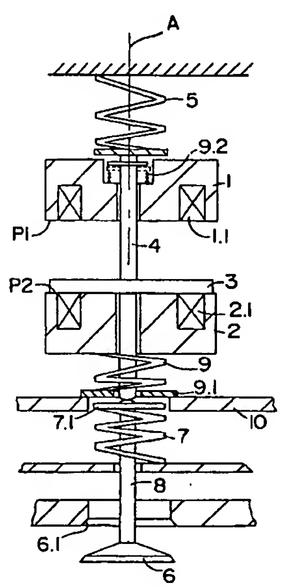
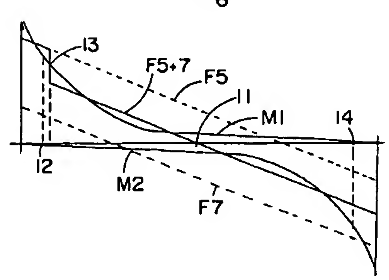
Attorney, Agent, or Firm—Spencer & Frank

[57] **ABSTRACT**

A method of operating a cylinder valve of an internal-combustion engine with an electromagnetic actuator for moving the cylinder valve into opposite open and closed valve end positions includes the steps of energizing an electromagnet of the actuator for generating an electromagnetic force for moving an armature of the actuator toward a pole face of the electromagnet against the force of a return spring; after the armature enters a zone adjacent the pole face of the electromagnet, applying to the armature an additional force opposing the electromagnetic force of the electromagnet; dimensioning the additional force such that an equilibrium between the electromagnetic force of the electromagnet and the increased opposing force of the return spring is situated at a location shortly before the armature enters into engagement with the pole face of the electromagnet; and after the armature reaches the location of force equilibrium, controlling a current supply to the electromagnet such that the armature reaches the pole face of the electromagnet with a predeterminable velocity.

10 Claims, 5 Drawing Sheets





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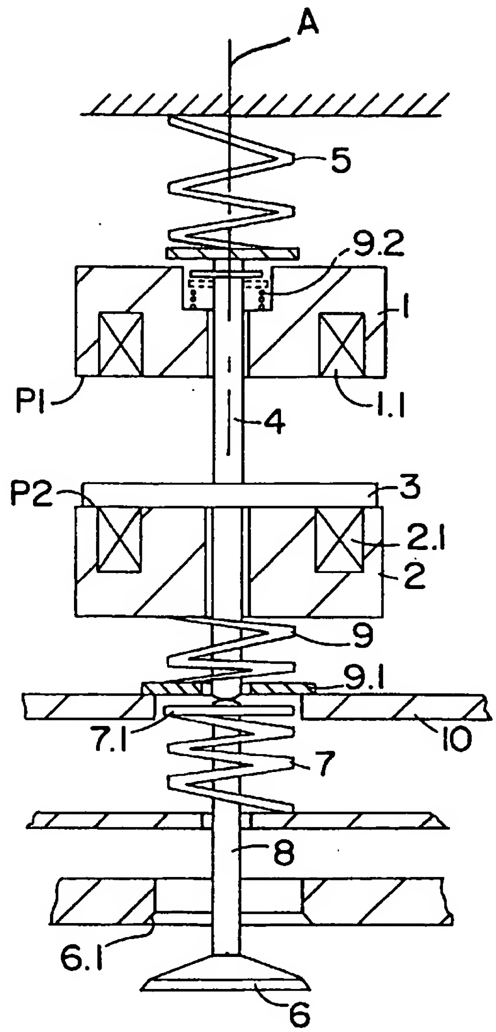

FIG. 1

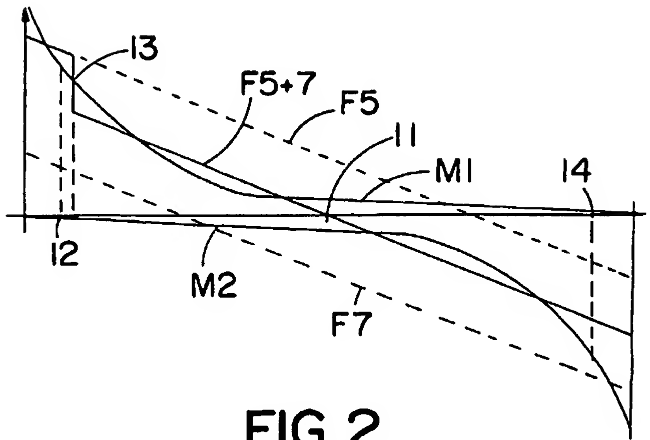

FIG. 2

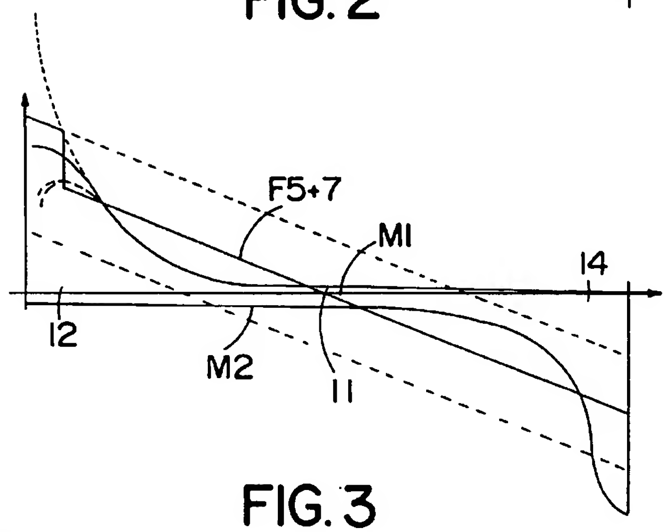

FIG. 3

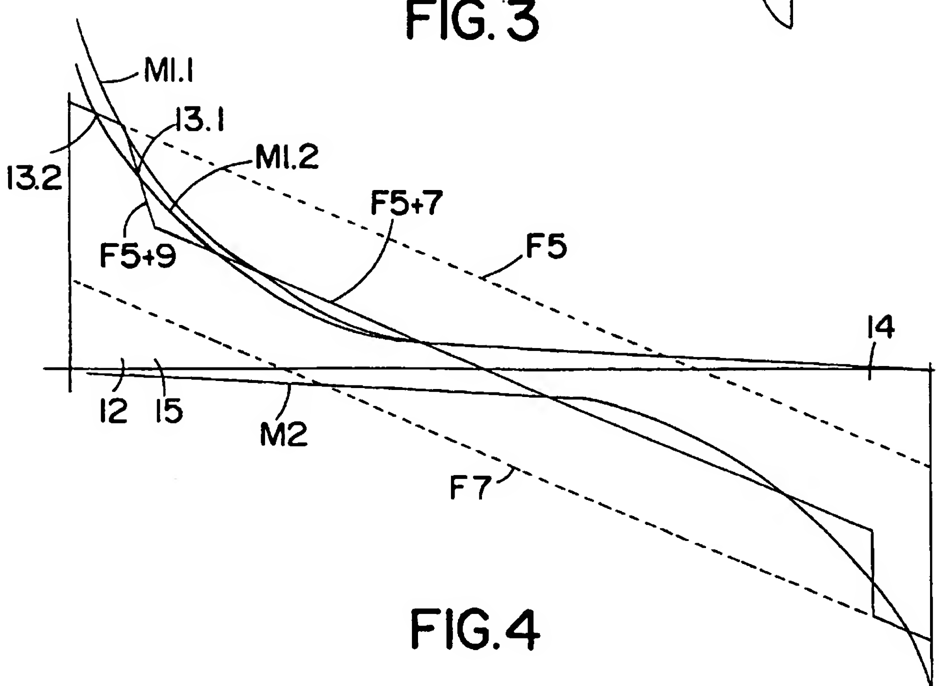

FIG. 4

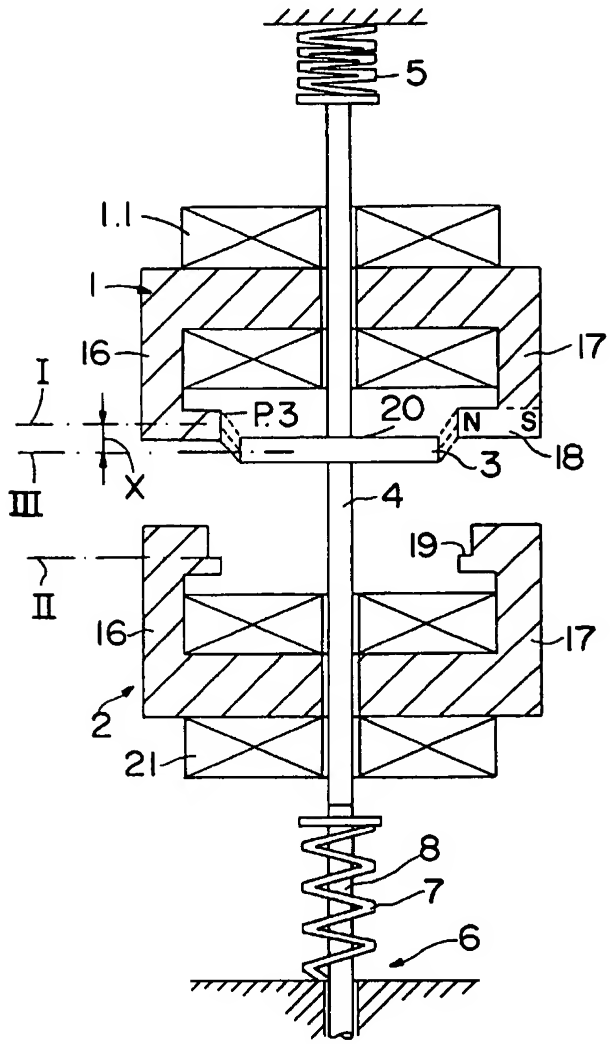

FIG. 5

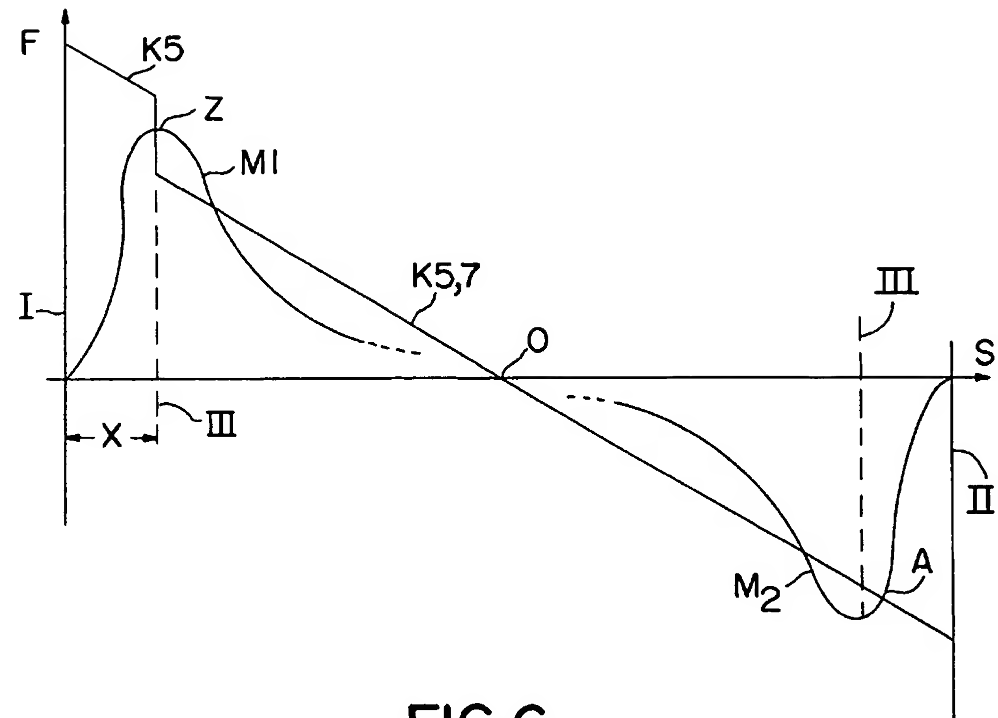

FIG. 6

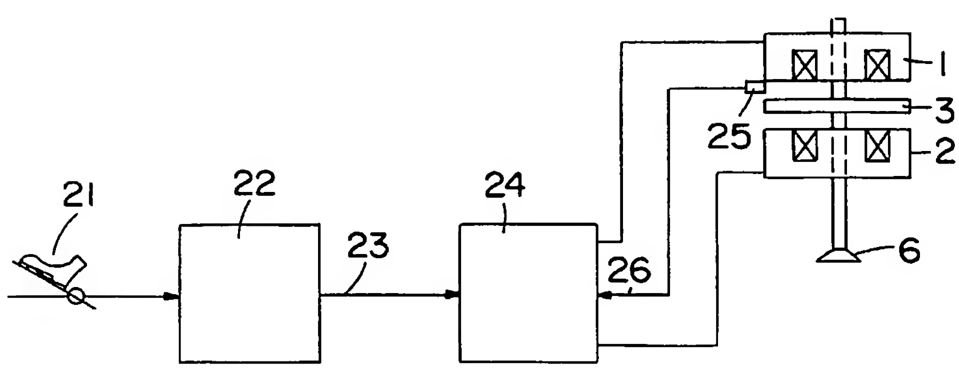

FIG. 7

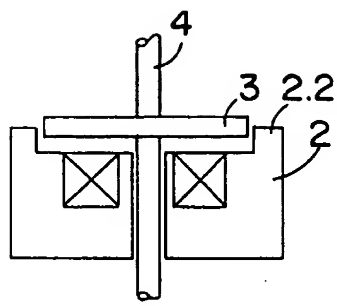

FIG.8.1

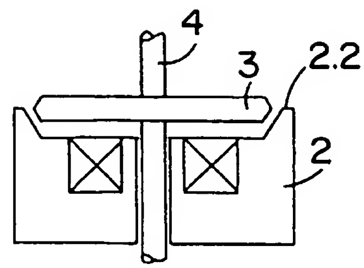

FIG.8.2

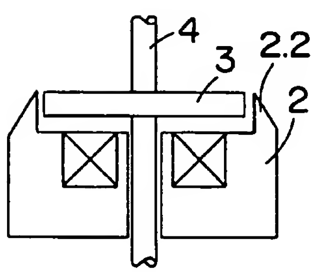

FIG.8.3

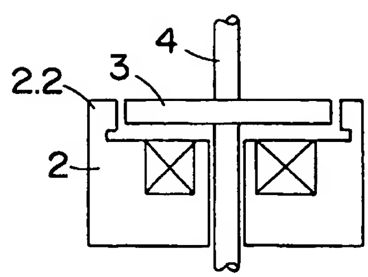

FIG.8.4

METHOD FOR CONTROLLING AN ELECTROMAGNETIC ACTUATOR OPERATING AN ENGINE VALVE

REFERENCE TO RELATED APPLICATION

This application claims the priority of German Application Nos. 196 51 846.6 filed Dec. 13, 1996 and 197 23 405.4 filed Jun. 4, 1997, which are incorporated herein by reference.

BACKGROUND OF THE INVENTION

U.S. Pat. No. 4,455,543 describes an electromagnetic actuator for operating a cylinder valve in an internal-combustion engine. The actuator essentially is formed of two spaced electromagnets wherein the pole faces of one electromagnet are oriented towards the pole faces of the other electromagnet. In the space defined between the pole faces of the two electromagnets an armature is disposed which is coupled with the cylinder valve and which, dependent upon the alternating energization of the electromagnets, moves back and forth against the force of resetting springs. In the respective terminal position of the cylinder valve (open or closed position) the armature is in engagement with the pole face of the respective holding magnet and is maintained there as long as the holding electromagnet is in an energized state. In the position of rest the armature is in a mid position between the pole faces of the two electromagnets, determined by the force equilibrium of the two oppositely working resetting springs.

If, during operation, the armature is to be displaced from one of its end positions into the other, the holding electromagnet is de-energized so that the armature, together with the cylinder valve, is moved by the force of the associated return spring in the direction of the mid position (position of rest). Upon subsequent energization of the opposite, capturing electromagnet, the armature will be in the effective range of the magnetic field of the capturing electromagnet and is moved by means of the magnetic force against the force of the other return spring into the other end position. As the armature impacts on the pole face of the capturing electromagnet, noise is generated whose intensity is dependent from the magnitude of the impact velocity. At the same time, risks are high that the armature rebounds which, for example, upon closing the cylinder valve, may lead to a brief reopening of the valve after the latter has already been seated. By a suitable control of the energization of the momentarily capturing electromagnet it is feasible to reduce the impact velocity; yet, the impact velocity has to be at all times sufficiently high to ensure that the armature is securely captured, that is, it securely arrives into engagement with the pole face.

To minimize the unavoidable impact noises, an impact velocity of less than 0.1 m/s is required for the armature in electromagnetic valve drives of the above-outlined type. Such low impact velocities have to be ensured in all real operating conditions while taking into consideration all related stochastic fluctuations; this, however, requires a substantial circuitry outlay. In vehicle engines irregularities in the road surface or other effects in the terminal approaching phase of the armature are sufficient to cause a sudden drop of the armature in case the magnetic force is oriented precisely in the direction of force requirement which is necessary for such a minimum impact velocity. A residual noise, however, cannot be avoided.

SUMMARY OF THE INVENTION

It is an object of the invention to provide an improved method for operating an electromagnetic valve actuator from which the discussed disadvantages are eliminated.

These objects and others to become apparent as the specification progresses, are accomplished by the invention, according to which, briefly stated, the method of operating a cylinder valve of an internal-combustion engine with an electromagnetic actuator for moving the cylinder valve into opposite open and closed valve end positions includes the steps of energizing an electromagnet of the actuator for generating an electromagnetic force for moving an armature of the actuator toward a pole face of the electromagnet against the force of a return spring; after the armature enters a zone adjacent the pole face of the electromagnet, applying to the armature an additional force opposing the electromagnetic force of the electromagnet; dimensioning the additional force such that an equilibrium between the electromagnetic force of the electromagnet and the increased opposing force of the return spring is situated at a location shortly before the armature enters into engagement with the pole face of the electromagnet; and after the armature reaches the location of force equilibrium, controlling a current supply to the electromagnet such that the armature reaches the pole face of the electromagnet with a predeterminable velocity.

Dependent upon design, the armature arrives into engagement with the pole face of the capturing electromagnet with a very small velocity which may be close to zero and upon such an arrival, the energizing current of the capturing electromagnet is increased to such an extent that a sufficient holding force is generated. According to another design, the armature does not directly impinge upon the pole face of the capturing electromagnet but is captured by a suitably directed magnetic field which is so positioned that the engine valve is, at least in its closed position, held by magnetic forces.

According to an advantageous feature of the invention, the force which augments the force of the return spring is applied by a spring force. Such an embodiment represents the simplest realization of the method according to the invention because it may be effected by mechanical systems, for example, by disconnecting the two resetting springs at least in the closed position of the valve, or by means of a resetting spring characteristic which at least in the end region is progressive or by applying the force of an additional spring. The application of an additional spring force has the advantage that in the vicinity of the pole faces the armature is exposed to an increasing, defined reinforcement of the resetting force of the resetting springs, so that in cooperation with the oppositely oriented magnetic forces, the point of equilibrium may be set to the open position and, in particular, to the closed position of the engine valve purely by a mechanical arrangement, and thus not only a soft arrival of the valve in its valve seat is ensured but also, the valve is securely held against dropping.

According to a further feature of the invention, the force which increases the force of the return springs is applied by magnetic forces affecting the armature. By influencing the magnetic forces in the vicinity of the pole faces, the position of the point of equilibrium can be controlled in a very accurate manner in coordination with the spring curve (which, at least in the end region has a progressive or abruptly changing course).

According to another advantageous feature of the invention, the force which augments the effect of the return springs and which is applied to the armature is derived from a suitably configured force-displacement curve of the electromagnetic forces generated by the capturing electromagnet. With this measure the armature may be brought in a "soft" manner to the pole faces of the capturing electromag-

net and may be maintained there. This may be achieved by a suitable control of the energization of the solenoid of the respective electromagnets and/or by a suitable design of the electromagnets.

In a further embodiment of the method according to the invention, for operating an engine valve which is designed for a soft arrival in its valve seat, the current supply to at least one of the two electromagnets of the electromagnetic valve actuator is so controlled and the generated electromagnetic field is so oriented that in an end position of the cylinder valve the reciprocatable armature which operates the valve is held at the electromagnet against a resetting spring force and out of contact with the pole faces of the electromagnet. In this method the armature does not impact on the pole face but is "softly" caught by an appropriately oriented magnetic field. By a suitable control of the energization of the electromagnet it is feasible to achieve a close-to-zero velocity as the armature reaches its end position. In case such an attempt is unsuccessful, the armature nevertheless may move without impacting the pole face, because the holding force and the positioning of the armature are effected exclusively by the magnetic field and the counterforce of the return spring. By virtue of an appropriate control of the energization (current supply) the magnetic field may have an excessive force which prevents an unintended "drop" of the armature in response to external force effects. Thus, even in case of a "creeping" of the armature into the end position, external impact forces cannot lead to a drop of the armature.

According to an advantageous feature of the invention, the magnetic field provided for a contactless holding of the armature is oriented substantially perpendicularly to the direction of armature motion, relatively to the end position of the armature. This is effected, for example, by providing that the respective electromagnet has two spaced pole faces which are essentially oriented towards one another. Upon release from the holding electromagnet the armature receives sufficient kinetic energy to move beyond the position of equilibrium determined by the resetting springs and it approaches the capturing electromagnet to such an extent that it arrives in the force range of the magnetic field of the capturing electromagnet and is thus continued to be moved in the direction of motion against the force of the effective return spring. As soon as the armature is situated between the two pole faces of the capturing electromagnet, the maximum magnetic force is exerted thereon so that the armature is maintained in the end position determined by those pole faces. The force of the magnetic field has to be designed such that it corresponds to the force of the compressed resetting spring.

According to a further advantageous feature of the method of the invention, upon energization of the electromagnet which holds the armature in a contactless manner, the current is reduced before the armature reaches its end position. In this manner it is possible to reduce the approaching velocity of the armature to avoid "overshooting". The energization may be controlled such that as the armature reaches its end position, the current is again increased to such an extent that the necessary holding force is exerted securely on the armature and thus an undesired armature motion triggered by outer force effects is avoided.

According to a further advantageous feature of the invention, the armature which is held in its end position in a contactless manner by the electromagnet, is additionally exposed to the force of at least one permanent magnet. The permanent magnet force is oriented in the same direction as the electromagnetic field and is of such a magnitude that it affects the armature less than the force of the associated return spring. By providing such a permanent magnetic field, the current required for holding the armature may be advantageously reduced particularly in the open position of the engine valve, because one part of the magnetic force is applied by the permanent magnet. To release the armature, the surplus force of the return spring is sufficient to overcome the force of the permanent magnet field after de-energization of the electromagnet.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic axial sectional view of an electromagnetic actuator for performing a preferred embodiment of the method according to the invention.

FIG. 2 is a diagram showing the magnetic and spring force curves as a function of the armature displacement in the method performed with the construction shown in FIG. 1.

FIG. 3 is a diagram showing the magnetic and spring force curves as a function of the armature displacement in another preferred embodiment of the method, including a control of the electromagnetic forces.

FIG. 4 is a diagram showing magnetic and spring force curves for different current intensities and using an additional spring.

FIG. 5 is a schematic axial sectional view of an electromagnetic actuator for performing another preferred embodiment of the method according to the invention, where the armature is positioned in a contactless manner.

FIG. 6 is a diagram showing the magnet and spring force curves as a function of the armature displacement for a method practiced with the actuator illustrated in FIG. 5.

FIG. 7 is a block diagram for performing the method according to the invention.

FIGS. 8.1–8.4 are schematic fragmentary side elevational views of various actuator structures illustrating different configurations of an electromagnet thereof.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

The electromagnetic actuator shown in FIG. 1 in a basic representation is formed essentially of two spaced electromagnets 1 and 2 functioning, respectively, as a closing magnet and an opening magnet. An armature 3 is arranged between the two electromagnets 1 and 2 for reciprocation parallel to the actuator axis A. The armature 3 is connected to a guide rod 4 which is guided in the region of the electromagnets 1 and 2.

A cylinder valve 6 operated by the electromagnetic actuator is maintained in the closed position by a closing spring 7. The cylinder valve 6 has a valve shaft 8 whose upper end is engaged by the bottom end of the guide rod 4. The upper end of the guide rod 4 is supported on a stationary component with the interposition of an opening spring 5. The opening spring 5 and the closing spring 7 affect the armature 3 via the guide rod 4 as resetting springs and exert oppositely oriented forces. When the electromagnets 1 and 2 are in a de-energized state, the armature 3 is maintained by the two resetting springs 5 and 7 in a position of equilibrium between the two electromagnets 1 and 2. By setting the bias of the opening spring 5 with the aid of non-illustrated setting means (such as a setscrew), the distance of the position of equilibrium from the two electromagnets 1 and 2 may be adjusted.

In the illustrated embodiment the solenoid 2.1 of the electromagnet 2 functioning as the opening magnet is

energized, as a result of which the cylinder valve 6 is maintained in its open position against the force of the closing spring 7 by the magnetic field of the electromagnet 2.

Upon de-energization of the electromagnet 2 and upon energization of the electromagnet 1 (simultaneously or with a suitable delay) functioning as the closing magnet, the armature 3 is released from the pole face P2 of the electromagnet 2 and moves, urged first by the force of the closing spring 7, up to the mid position and thereafter, urged by the accelerating forces and the magnetic forces of the electromagnet 1, the armature 3 continues to move in the direction of the pole face P1 of the electromagnet 1.

The return spring 7 loses its force effect on the armature 3 as soon as the engine valve 6 assumes its closed position and thus the valve head engages the valve seat 6.1. Then the armature 3, by virtue of its kinetic energy and under the effect of the magnetic field moves against the force of the resetting spring 5 through a small distance corresponding to the axial clearance between the guide rod 4 and the valve stem 8, until it reaches its final position of engagement at the pole face P1. The course of the forces is illustrated in FIG. 2 where the magnet force of the electromagnet 1 is designated at M1, the force of the electromagnet 2 is designated at M2, the force of the opening spring 5 is designated at F5 and the force of the closing spring 7 is designated at F7. The spring characteristic curve F5+7 shows the resulting force of the two return springs 5 and 7. The curve portion 13 designates the "force jump" upon closing the engine valve.

As it has been described earlier in conjunction with FIG. 1, upon de-energization of the electromagnet 2 the armature 3 is first exposed practically only to the resetting force F7 of the closing spring 7. When the armature 3 passes through the location 11 where between the force F7 of the closing spring 7 and the force F5 of the opening spring 5 an equilibrium prevails, the kinetic energy drives the armature 3 beyond the location 11. Since in the meantime the capturing electromagnet 1 is energized, the armature 3 arrives in the zone of the magnetic field (which is under build-up) and is, by means of the magnetic force M1 moved further in the direction of the pole face P1 of the capturing electromagnet 1 against the resetting force F5 of the resetting spring 5. The exponentially increasing magnet force M1 accelerates the armature 3 until it impacts on the pole face P1 at location 12.

As may be observed in FIG. 2, the course of the forces affecting the armature 3 indicates that shortly before the armature 3 engages the pole face P1 at location 13, a position of equilibrium is reached between the magnetic force M1 and the spring force F5 of the return spring 5 which is the only effective spring at that time.

As shown in FIG. 3 for the motion into the "valve closed" position, in the zone of the final armature approach to the pole face—in the present instance the pole face P1—the magnetic force pattern may be affected by control measures or by a suitable structural design of the capturing electromagnet and/or the armature 3. The dotted continuation of the curve M1 shows the magnetic force curve without saturation. The solid-line portion of the curve M1 in the region of the closed position shows, in turn, the course of the magnetic force having substantial saturation. The dash-dot portion of the curve M1 illustrates the course of the magnetic force when the energizing current is reduced in the region of the equilibrium position. This circumstance indicates that even in conventional electromagnetic actuators for cylinder valves a soft arrival of the armature at the pole face of the respective electromagnet may occur.

To ensure that in the course of the motion behavior as discussed above, the cylinder valve 6 engages the valve seat 6.1 in a soft manner, the actuator shown in FIG. 1 is provided with an additional spring 9 which engages, with the interposition of a spring seat disk 9.1, a supporting surface 10 forming part of a housing not shown in detail. The arrangement is designed such that upon the closing motion of the cylinder valve 6 the spring seat disk 7.1 of the closing spring 7 abuts the spring seat disk 9.1 of the additional spring 9 shortly before the cylinder valve 6 arrives into engagement with the valve seat 6.1.

Since upon motion of the armature 3 beyond the position of equilibrium the resetting force of the opening spring 5 becomes effective to oppose the magnetic force of the electromagnet 1 with a spring force determined by its spring characteristic, additionally to the resetting force of the return spring 5 the armature 3 is exposed to the force effect of the additional spring 9. As a result, a defined additional force effect is present which opposes the magnetic force while the latter progressively increases as the armature approaches the pole face.

As shown in FIG. 4, shortly before reaching the location 12, however, the valve 6 becomes operatively coupled to the spring 9 so that in addition to the resetting force F the armature 3 is exposed to the defined force component of the additional spring 9 in the opposite direction. Thus, shortly before the arrival of the valve 6 into the valve seat 6.1, that is, shortly before reaching the location 12, the armature 3 passes through a location 13 at which a force equilibrium between the magnetic force M1 on the one hand and the force F5+9 composed of the spring forces F5 and F9 prevails. By the defined force effect the armature 3 is braked in its motion.

The armature 3, for its displacement from the location of equilibrium 13 to its eventual closed position, has to be guided by a positive control of the current supplied to the capturing electromagnet 1 as it has been generally described in detail above.

If the capturing electromagnet 1 is energized with a higher current intensity, there is obtained a magnetic force curve M1.1. In this case the energizing current is so controlled that the point of the force equilibrium is identical with the location of the closed position of the valve.

If, however, the electromagnet 1 is energized with a lesser current as shown by the curve M1.2, there are obtained two stable points of equilibrium, that is, the points 13.1 and 13.2. It is seen from the spring force curve and the magnet force curve in the region between points 15 and 12, particularly between points 13 and 12 that by means of controlling the energization of the electromagnet 1 a positive guidance of the armature motion is feasible.

In the embodiment shown in FIG. 1, essentially one additional spring 9 is provided for the valve motion into the closed position so that, as shown in FIG. 2, upon motion into the open position, the conventional impacting conditions occur, that is, the armature 3, upon reaching the "valve open" position, impacts hard on the pole face P2 of the electromagnet 2 due to the large excess of the magnetic force M2 with respect to the force F7 of the resetting spring 7. If such an occurrence is to be avoided, the opening spring 5 too, is associated with an additional spring 9.2 as shown in dash-dot lines in FIG. 1. Since during the opening motion the only impacting occurs between the armature 3 and the pole face P2, the additional spring 9.2 may be designed differently from the additional spring 9.

While, as described above, the method practiced with the electromagnetic actuator of FIG. 1 applies an additional

force by the spring **9**, that is, by mechanical means, in the construction according to FIG. **5** the additional force is generated by magnetic means. The basic construction of the electromagnetic actuator of the embodiment shown in FIG. **5** generally corresponds to that of the actuator shown in FIG. **1**.

The two electromagnets **1** and **2** are substantially of identical construction; they each have two lateral pole shoes **16** and **17** whose pole faces (such as P3) are oriented in a direction perpendicular to the actuator axis A, so that when a magnetic field is generated by the energization of the respective electromagnet, the magnetic field is oriented substantially perpendicularly to the actuator axis A and thus to the direction of motion of the armature **3**. The resetting force of the opening spring **5** on the one hand and the force of the magnetic field of the electromagnet **1** on the other hand are so dimensioned that in the shown closed position the armature **3** assumes a position in which it is situated slightly lower than the height level of the two pole faces **11**. As shown by the symbolically illustrated field lines, in the illustrated position a magnetic residual force remains which acts in the closing direction; the force equilibrium is, however, so selected that the lower end of the guide rod **4** does not lift off the upper end of the valve shaft **8**.

Upon de-energizing the coil **1.1** of the electromagnet **1**, the armature **3** moves under the force of the compressed opening spring **5** towards the electromagnet **2**. Upon energization of the coil **2.1** of the electromagnet **2**, for example, at the moment when the armature traverses the mid position, the armature **3** is exposed to the magnetic field of the electromagnet **2** and is, as a result, pulled into the "valve open" position in between the two pole faces of the electromagnet **2** and is held in that position against the force of the closing spring **7**, so that the cylinder valve **6** is maintained in its open position.

For supporting the electromagnetic holding force, a permanent magnet may be arranged at the pole shoe **16** and/or **17** of the electromagnets **1** and/or **2** and co-oriented with the electromagnetic field of the energized electromagnets **1** and **2**. FIG. **5** shows one permanent magnet **18** attached to the pole face of the pole shoe **17** of the electromagnet **1** and forming the radially inwardly directed leg thereof. By virtue of the magnetic force of such permanent magnet or magnets, the magnetic field of the electromagnets **1** and/or **2** is reinforced in the holding position, so that a lower current intensity is required for holding the armature **3**.

As seen in FIG. **5**, the armature **3** may move further beyond its respective end position without impacting the pole face, in case the velocity of the armature is greater than zero when the end position is reached. The flat force pattern achieved by the orientation of the magnetic field in the vicinity of the desired end position permits a soft armature approach which may be controlled by regulating the current of the capturing electromagnet.

In the illustrated embodiment the armature **3** is a circular disk-shaped component and the pole faces P3 are of hollow cylindrical shape. The air gap between the outer periphery of the armature **3** and the pole faces P3 is 0.1 mm at the most. Such a small air gap ensures that the required holding current may be maintained at a practical magnitude. By virtue of the circular design of the armature **3** it is ensured that the torsional armature motions caused by the return springs **5** and **7** have no effect on the operation and free mobility of the armature **3**. By arranging a step **19** on the pole faces, one part of the magnetic field may be oriented in a direction towards the mid position so that the capturing magnetic force has an earlier effect on the armature **3** as it approaches the capturing electromagnet **2**.

The actuator illustrated in FIG. **5** may be modified such that the electromagnet which holds the armature **3** in the closed position has a conventional construction, that is, its pole face is oriented towards the surface **20** of the armature **3** so that—as heretofore—the armature is held in contact with the pole face of the electromagnet. Although noise generation upon impacting of the armature on the pole face will occur, the required holding current, however, may be reduced.

FIG. **6** illustrates the forces F acting on the armature **3** as a function of the armature displacement s. The curve M_1 represents the magnetic force of the electromagnet **1** from the effective plane I toward the armature **3**, while the curve M_2 represents the magnetic force of the electromagnet **2** from the effective plane II, in the opposite direction toward the armature **3**. Starting from the position of equilibrium **0** of the forces of the return springs **5** and **7**, directed oppositely to one another across the point of equilibrium **0**, on opposite sides of the abscissa s a practically linear spring characteristic $K_{5,7}$ is obtained which exerts a force on the armature **3** dependent upon its momentary position from the point of equilibrium **0**.

In the "valve closed" position of the closing magnet **1** the characteristic curve for the spring force has a "jump" caused by the fact that after the valve **6** is seated, only the opening spring **5** can act on the armature **3**, while the closing spring **7** holds the valve **6** firmly against its seat.

A comparison between the magnet force curves on the one hand and the spring force curves on the other hand show that a position of equilibrium is given for the "valve closed" position at the peak Z of the magnet force curve M_1 and for the "valve open" position in the point of intersection A between the spring curve K and the magnet force curve M_2. The distance between the two positions of equilibrium equals the valve stroke. The distance x between the central plane III of the armature shown in FIG. **5** and the central plane I of the pole faces of the electromagnet **1** indicates the position of the armature **3** in the position of equilibrium in the "valve closed" position. Likewise, the "valve open" position is determined by the distance of the mid plane III of the armature **3** to the mid plane II of the pole faces of the electromagnet **2**.

It is seen from FIG. **6** that in the region of the point Z and the point A the magnet force is greater than the spring force so that a displacement of the armature from either end position under the effect of external impact forces is practically not possible and therefore an unintentional closing or opening of the cylinder valve in response to such external forces is reliably prevented.

By means of a suitable control of the current supply to the electromagnets **1** and **2** the course of the curves M_1 and M_2 may be altered.

FIG. **7** shows a block diagram illustrating the control of the electromagnetic actuators in a piston-type internal-combustion engine. For the sake of simplicity the circuit is shown only for a single one of a plurality of actuators present in the engine. The engine is controlled conventionally by a driver-operated gas pedal **21** whose setting signal is applied to an electronic engine control apparatus **22** by means of which, dependent upon the momentary operational data (such as engine rpm, coolant temperature, etc.), the individual functions of the engine are conventionally controlled, including the operation of the engine valves. For this purpose control signals **23** are applied by electronic

engine control apparatus **22** to a current regulator **24** which controls the electromagnets **1** and **2** of the electromagnetic actuators for operating the engine valves **6**.

In the described embodiment a path sensor **25** is associated at least with the electromagnet **1** operating as the closing magnet for detecting the approach of the armature **3** to the pole face **P1**, so that by means of a corresponding signal **26** the earlier-described current regulation may be effected in order to bring the armature **3** smoothly into engagement with the pole face **P1** from the point of the equilibrium between the resetting forces and the magnetic force.

FIGS. **8.1** and **8.2** schematically show different magnet configurations by means of which the position of the force equilibrium for the armature may be affected. While according to the embodiment shown in FIG. **5** the armature does not engage the pole faces, the configurations in FIGS. **8.1** and **8.2** are so designed that the magnetic forces in the last phase of approach to the pole face allow an increase of influence of the force, so that by means of a positive guidance of the current flow through the capturing electromagnet a soft but secure arrival of the armature on the pole face is ensured.

FIGS. **8.3** and **8.4** show magnet configurations in which a force pattern is achieved which, when the armature arrives in the immediate vicinity of the pole faces, results in a weakening of the force. In this manner a stable position of equilibrium without additional springs may be achieved in the vicinity of the pole faces. In the magnet configurations of FIGS. **8.1–8.4** the desired effect is achieved in each instance by variously configured pole shoe "horns" **2.2** projecting beyond the pole faces.

It will be understood that the above description of the present invention is susceptible to various modifications, changes and adaptations, and the same are intended to be comprehended within the meaning and range of equivalents of the appended claims.

What is claimed is:

1. A method of operating a cylinder valve of an internal-combustion engine with an electromagnetic actuator for moving the cylinder valve into opposite open and closed valve end positions; the method comprising the steps of

(a) energizing an electromagnet of the actuator for generating an electromagnetic force for moving an armature of the actuator toward a pole face of the electromagnet against the force of a return spring;

(b) after the armature enters a zone adjacent the pole face of the electromagnet, applying to the armature an additional force opposing the electromagnetic force of the electromagnet;

(c) dimensioning the additional force such that an equilibrium between the electromagnetic force of the electromagnet and the increased opposing force of the return spring is situated at a location shortly before the armature enters into engagement with the pole face of the electromagnet; and

(d) after the armature reaches the location of force equilibrium, controlling a current supply to the electromagnet such that the armature reaches the pole face of the electromagnet with a predeterminable velocity.

2. The method as defined in claim **1**, wherein step (b) comprises the step of applying a spring force as said additional force.

3. The method as defined in claim **2**, wherein step (b) comprises the step of applying a spring force of an additional spring.

4. The method as defined in claim **2**, wherein step (b) comprises the step of applying a spring force of return spring having a progressive spring characteristic curve at least in an end region of a motion path of said armature.

5. The method as defined in claim **1**, wherein step (b) comprises the step of applying magnetic forces to said armature.

6. The method as defined in claim **1**, wherein step (b) comprises the step of applying magnetic forces to said armature by shaping a force/displacement course of magnetic forces generated by said electromagnet.

7. The method as defined in claim **1**, wherein said electromagnet is a holding magnet for maintaining the cylinder valve in the closed position; further wherein step (a) comprises the steps of directing the electromagnetic field of said electromagnet and controlling the current supply to said electromagnet such that in the closed end position of the cylinder valve the armature is held in the armature end position at said electromagnet against the spring force and out of contact with the pole faces of said electromagnet.

8. The method as defined in claim **7**, wherein the step of directing the electromagnetic field comprises the step of directing the electromagnetic field substantially perpendicularly to the direction of the armature motion.

9. The method as defined in claim **7**, wherein said step of controlling the current supply comprises the step of reducing the current supplied to said electromagnet shortly before the armature reaches the armature end position at said electromagnet.

10. The method as defined in claim **7**, further comprising the steps of directing the magnetic force of a permanent magnet to said armature in the armature end position thereof at said electromagnet codirectionally with the electromagnetic force of one electromagnet and dimensioning the magnetic force of said permanent magnet such as to be less than the spring force opposing the electromagnetic force applied to said armature in said armature end position.

* * * * *

US006076490A

United States Patent [19]

Esch et al.

[11] **Patent Number: 6,076,490**

[45] **Date of Patent: Jun. 20, 2000**

[54] **ELECTROMAGNETIC ASSEMBLY WITH GAS SPRINGS FOR OPERATING A CYLINDER VALVE OF AN INTERNAL-COMBUSTION ENGINE**

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[73] Assignee: **FEV Motorentechnik GmbH & Co.KG**, Aachen, Germany

[21] Appl. No.: **09/123,987**

[22] Filed: **Jul. 29, 1998**

[30] **Foreign Application Priority Data**

Jul. 31, 1997 [DE] Germany 197 33 186

[51] **Int. Cl.**^{7} **F01L 9/04**

[52] **U.S. Cl.** **123/90.11**; 123/90.65; 251/129.1; 251/129.15; 251/129.18

[58] **Field of Search** 123/90.11, 90.14, 123/90.65; 251/129.01, 129.02, 129.1, 129.15, 129.16, 129.18

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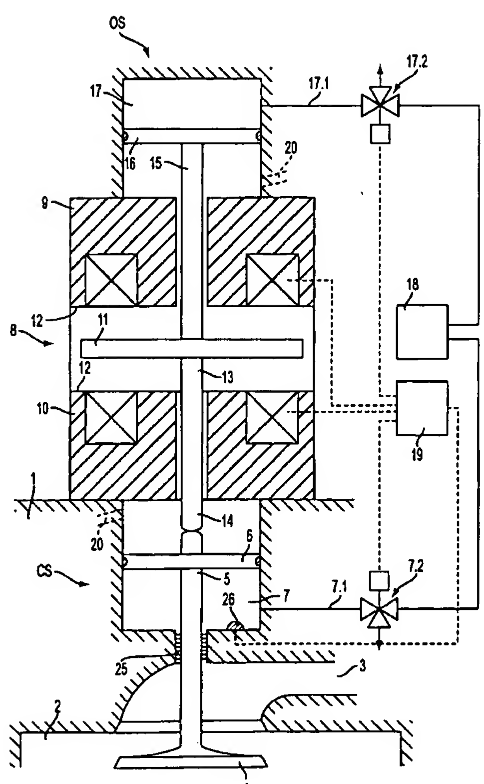
Primary Examiner—Wellun Lo
Attorney, Agent, or Firm—Venable; Gabor J. Kelemen

[57] **ABSTRACT**

A combination of a cylinder valve of an internal-combustion engine with an electromagnetic assembly for operating the valve. The electromagnetic assembly includes an electromagnetic actuator having first and second electromagnets supported in a spaced relationship with respect to one another, an armature movable between the first and second electromagnets and connected to the valve for moving it between open and closed positions. The electromagnetic assembly further includes a first control device for energizing the electromagnets to cause a motion of the valve by electromagnetic forces generated by the first and second electromagnets; and first and second pressurizable resetting gas springs operatively coupled to the armature to oppose motions thereof caused by the electromagnetic forces generated by the first and second electromagnets, respectively. A pressure supply device supplies pressure to the first and second resetting gas springs. A second control device controls the pressure supplied to the first and second resetting gas springs.

4 Claims, 3 Drawing Sheets



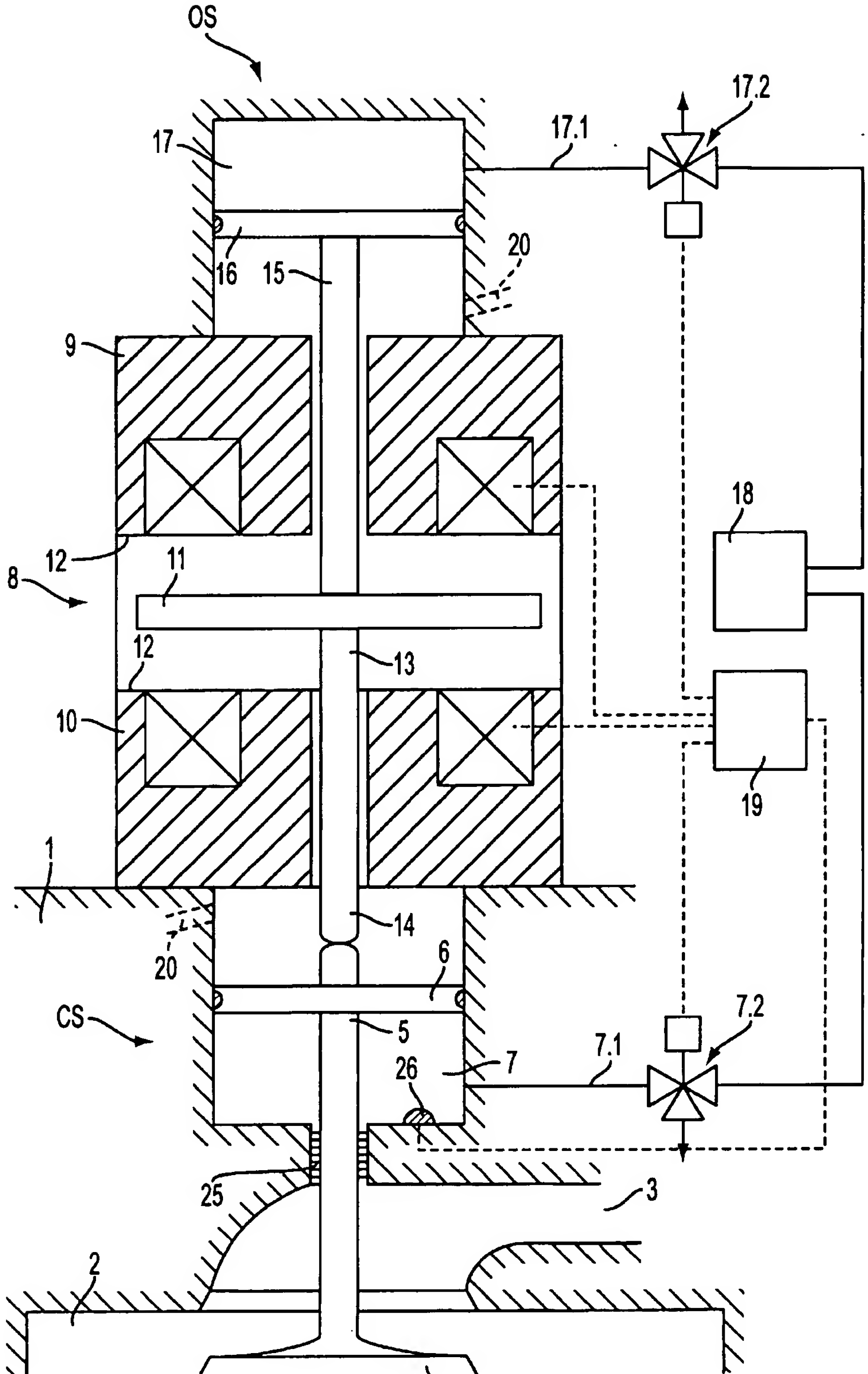

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FIG. 1

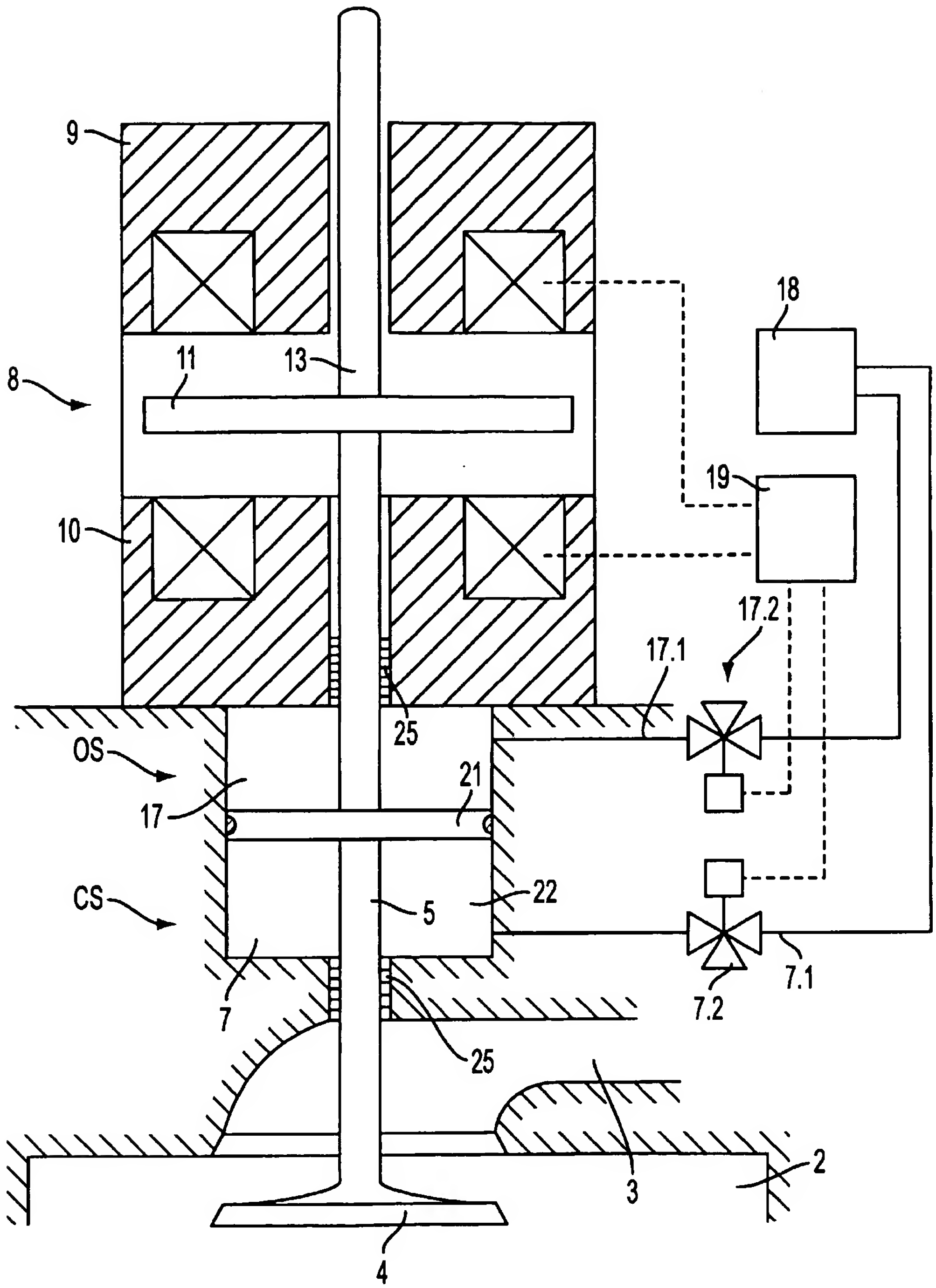

FIG. 2

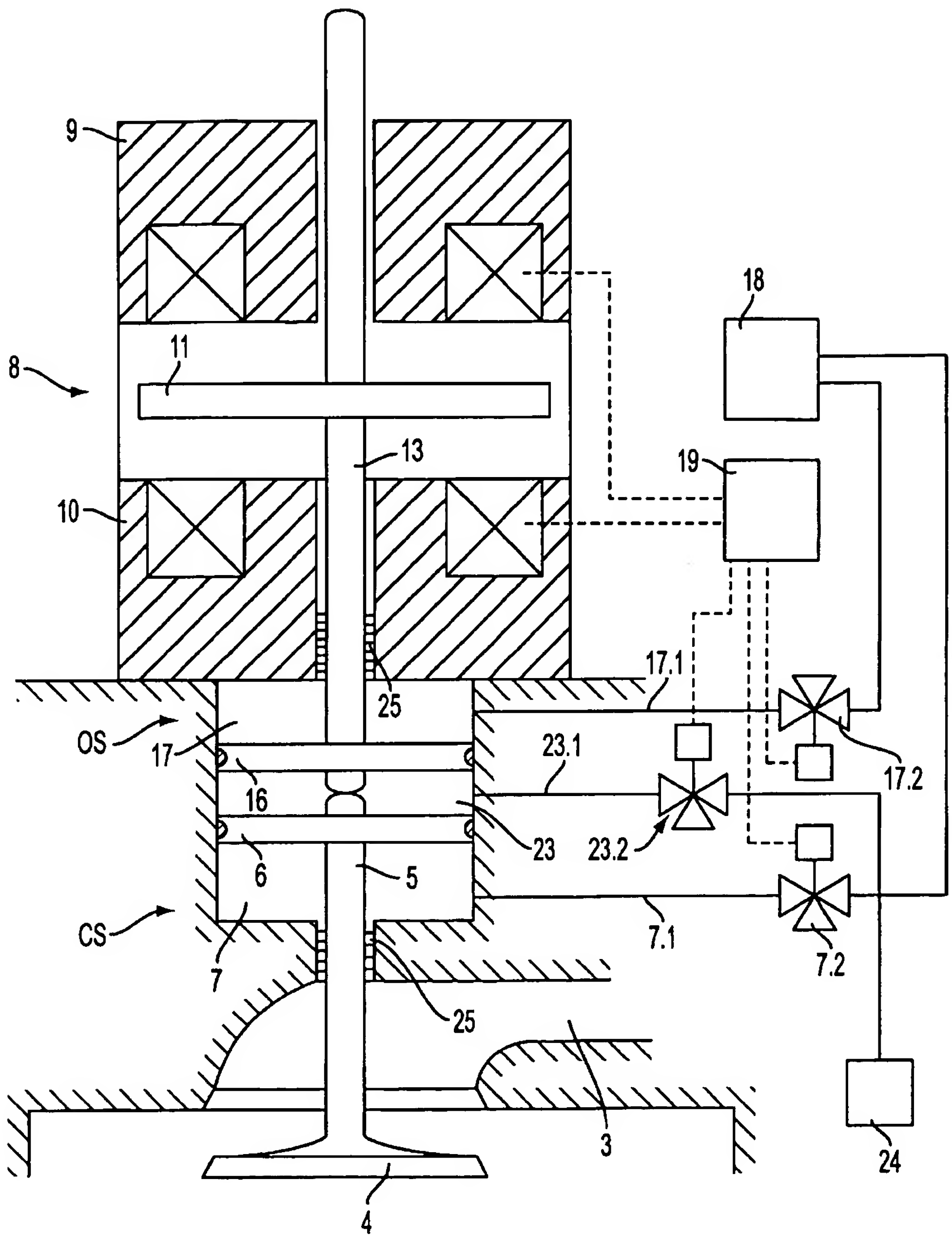

FIG. 3

ELECTROMAGNETIC ASSEMBLY WITH GAS SPRINGS FOR OPERATING A CYLINDER VALVE OF AN INTERNAL-COMBUSTION ENGINE

CROSS REFERENCE TO RELATED APPLICATION

This application claims the priority of German Application No. 197 33 186.6 filed Jul. 31, 1997, which is incorporated herein by reference.

BACKGROUND OF THE INVENTION

In piston-type internal-combustion engines the cylinder valves have been controlled until recently by a cam shaft of the engine with the intermediary of a linkage system mechanically coupling the cam shaft with the cylinder valve. As an alternative to such a valve control, in more recent engine designs electromagnetic actuators have been used for operating the cylinder valves. Such an electromagnetic actuator includes two spaced electromagnets between which an armature reciprocates in response to electromagnetic forces generated as a function of the energizing currents controlled by an electronic control system of the engine. The armature, in response to electromagnetic forces, moves against the force of resetting springs and is coupled with the respective cylinder valve to effect corresponding motions thereof. Such a system is described, for example, in U.S. Pat. No. 4,455,543.

Heretofore mechanical springs such as coil springs have been used as resetting springs, which, in principle, have been found to be satisfactory.

Operating a cylinder valve with an electromagnetic actuator controlled by an electronic control system makes possible a freely variable valve control, that is, it is feasible to vary both the opening moment and the open period of the valve as a function of the load requirements of the engine. In the design of the electromagnetic actuator, however, the oscillation characteristics of the spring/mass system composed of the armature and the cylinder valve as the mass and the mechanical resetting springs as the spring element have to be considered as a fixed, given magnitude.

SUMMARY OF THE INVENTION

It is an object of the invention to provide an electromagnetic valve-operating assembly of the above-outlined type, whose adaptability to the operating conditions is further improved.

This object and others to become apparent as the specification progresses, are accomplished by the invention, according to which, briefly stated, the electromagnetic assembly for operating a cylinder valve of an internal-combustion engine includes an electromagnetic actuator having first and second electromagnets supported in a spaced relationship with respect to one another and an armature movable between the first and second electromagnets and connected to the valve for moving it between open and closed positions. The electromagnetic assembly further includes a first control device for energizing the electromagnets to cause a motion of the valve by electromagnetic forces generated by the first and second electromagnets; and first and second pressurizable resetting gas springs operatively coupled to the armature to oppose motions thereof caused by the electromagnetic forces generated by the first and second electromagnets, respectively. A pressure supply device supplies pressure to the first and second resetting gas springs. A

second control device controls the pressure supplied to the first and second resetting gas springs.

The use of gas springs in which the gas pressure applied thereto and thus the spring constant is variable, is advantageous, because by varying the spring constant with the otherwise present engine control system, the spring constants and thus the oscillation characteristics of the oscillating system composed of the armature, the cylinder valve and the resetting gas springs may be adapted to the momentary load conditions. The resetting force and thus the oscillating characteristics may be adapted to the momentary operating conditions of the engine by suitably increasing or reducing the gas pressure applied to the gas spring, based on an initial, predetermined resetting force. An increase of the resetting force is, for example, expedient in a high-rpm operation to effect the high acceleration of the armature and the cylinder valve necessary to achieve short operating periods.

By positioning the resetting gas springs externally of the electromagnets, the external dimensions of the electromagnets, on the one hand, and the dimensions of the gas springs, on the other hand, may be optimally adapted to the existing requirements. As an eventual result, systems of narrow constructions may be built which can be accommodated in the limited space available above a cylinder of an internal-combustion engine. This advantage is of particular significance in engines in which each cylinder is provided with two intake valves and two exhaust valves.

According to the invention, one gas spring is connected with the cylinder valve and acts as a closing spring while the other gas spring is connected with the armature and acts as an opening spring. The armature, with its guide and the cylinder valve may form a closed, one-piece structural unit. It is, however, expedient to provide that the armature, together with its opening spring and the cylinder valve, together with its closing spring are movable independently from one another. By means of such a separation of the two structural components from one another, a wear compensation is feasible in the region of the valve seat and in the system itself by means of suitable automatic valve clearance (valve slack) adjusting devices. Such wear appears due to the different heat expansions in response to different temperatures particularly at the stem of the cylinder valve.

According to a particularly advantageous feature of the invention, the opening gas spring and the closing gas spring are arranged on the same side of the two electromagnets viewed as one assembly. This arrangement provides the possibility to manufacture the electromagnetic part, on the one hand, and the gas pressure-affected part, on the other hand, as a closed prefabricated unit which may be subsequently assembled to form the final aggregate.

While it is feasible in principle to use any system as a gas spring which makes possible a change of the gas pressure during operation (for example, a gas spring which operates similarly to a gas bubble storing device), according to a particularly advantageous feature of the invention, each gas spring is composed of a piston-and-cylinder unit. It is particularly expedient to guide the two gas springs (opening and closing springs) in a single, common cylinder.

According to another advantageous feature of the invention, the common cylinder accommodates two pistons which form part of the opening gas spring and the closing gas spring, and which are arranged at a distance from one another. The space defined between the two pistons may be separately pressurized in a controlled manner from a separate pressure source. By virtue of this arrangement, when the

armature is held in the closed position by the electromagnetic actuator, an opening of the cylinder valve to execute a small stroke may be effected by pressurizing the intermediate space without energizing a magnet of the electromagnetic actuator. The sole condition to effect such an operation is to pressurize the intermediate space with a pressure medium, preferably a pressurized gas, with a pressure which is greater than that of the closing gas spring. By means of suitably controlling the magnitude and duration of the pressure, a desired partial opening stroke of the cylinder valve may be accordingly obtained.

The above-outlined construction permitting a partial opening of the cylinder valve makes possible an operation where, for example, the gas intake valve, during a suction cycle, is opened only for a short period with only a partial stroke to thus present an only partially open cross-sectional flow passage area. As a result, the fresh combustion gas flows into the cylinder at a high velocity, generating a vortex in the combustion chamber. Following the subsequent full opening of the cross-sectional flow passage area of the valve, such a vortex improves the mixture formation and the motion of the charge in the cylinder. Since in such a case too, the pressurizing of the intermediate space between the two gas spring pistons may be freely controlled, such an operation may be practiced between wide limits.

According to a further advantageous feature of the invention, the pressure chamber of at least one gas spring is provided with a pressure sensor which is connected with a control device for determining the position of the armature. By setting the system for a predetermined mode of operation such that in the de-energized state of the electromagnets the armature is in a mid position between the two electromagnets, in which case the gas pressures in the closing gas spring and in the opening gas spring are also identical (provided that the chamber volumes and the piston faces are identical), an operation is obtained where an armature motion against the resetting force of one gas spring causes an increase of the gas pressure therein and a decrease of the gas pressure in the opposite gas spring. The change of the gas pressure is therefore proportional to the position of the armature between the two electromagnets and thus information concerning the position of the armature may be obtained by detecting the pressure change. The signal obtained from the pressure sensor may then be applied to the control device. This is of significance particularly as the armature approaches the capturing electromagnet because in such a case the control device may accordingly control the current supply to the capturing electromagnet.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic axial sectional view of a preferred embodiment of the invention.

FIG. 2 is a schematic axial sectional view of a further preferred embodiment of the invention.

FIG. 3 is a schematic axial sectional view of yet another preferred embodiment of the invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

FIG. 1 illustrates one part of a cylinder head **1** of a piston-type internal-combustion engine in the port region of an engine cylinder **2**. The port **3** which opens into the cylinder **2** and which may be an intake port or an exhaust port, may be opened or closed by a cylinder valve **4** in accordance with a control based on the operating cycle of the engine. The cylinder valve **4** has a valve stem **5** carrying a piston **6** which is accommodated for axial sliding motion in a cylinder **7** provided in the cylinder head **1**.

The cylinder valve **4** is operated by an electromagnetic actuator **8** which is essentially composed of a closing magnet **9** and an opening magnet **10** situated at a distance from the closing magnet **9**. Between the two magnets **9** and **10** an armature **11** is guided for a reciprocating motion. In the embodiment illustrated, the armature **11** is situated, in the de-energized state of the electromagnets **9** and **10**, midway between the pole faces **12** of the two magnets **9** and **10**.

The armature **11** is affixed to a guide rod **13** which, at its free end **14**, is connected with the valve stem **5** and which, at its other free end **15** is provided with a piston **16** guided in a cylinder **17**. The cylinders **7** and **17** are connected with a pressure supply source **18** by means of respective pressure lines **7.1** and **17.1**. In the lines **7.1** and **17.1** respective controllable valves **7.2** and **17.2** (shown as three-way valves) are disposed, so that dependent on the position of the valve **5**, the pressure in the cylinders **7** and **17** may be increased or reduced. The setting drives for the valves **7.2** and **17.2** are connected with a control device **19** forming an integral part of the engine control system.

The control device **19** also controls the current supply to the electromagnets **9** and **10** dependent on operational requirements of the engine. The cylinder **7** together with the piston **6** and the cylinder **17** together with the piston **16** each constitute a gas spring which serves in the described embodiment as resetting springs for the electromagnetic system. The piston-and-cylinder unit **6, 7** thus forms a closing gas spring CS and the piston-and-cylinder unit **16, 17** constitutes an opening gas spring OS for the cylinder valve **4**. If the two gas springs CS and OS are charged with a predetermined (for example, identical) pressure, the armature **11**, in case of a de-energized state of the electromagnets **9** and **10**, assumes its illustrated position midway between the pole faces **12** of the electromagnets **9** and **10**. As the armature **11**, for example, by means of particular starting measures, arrives into engagement with the closing magnet **9** upon energization thereof, the pressure in the opening spring OS accordingly increases. Upon de-energizing the closing magnet **9** the opening spring OS accelerates the armature **11** in the direction of the opening magnet **10** and as the armature **11** passes its mid position, the gas pressure in the closing spring CS increases as the distance between the approaching armature **11** and the pole face **12** of the opening magnet **10** decreases. As the armature **11** traverses its mid position, the opening magnet **11** is energized so that the generated electromagnetic field captures the armature **11** and carries it into engagement with the pole face **12** of the opening magnet **10**, whereupon the cylinder valve **4** is held in its open position in accordance with a period of energization determined by the control device **19**. Upon closing of the cylinder valve **4**, the opening magnet **10** is de-energized and the closing magnet **9** is energized in a reverse sequence. The reciprocating motion of the armature **12** and the cylinder valve **4** resulting from such a current control thereafter progresses in accordance with cycles inputted into the engine control system by the engine rpm. The work chambers of the cylinders **7** and **17** oriented towards the electromagnets **9** and **10** have respective vents **20** to avoid an adverse effect on the motion characteristics of the armature **11** in case an encapsulated magnetic system is used.

The embodiment illustrated in FIG. **2** corresponds in its principal construction and mode of operation essentially to the embodiment illustrated in FIG. **1**. In the FIG. **2** embodiment, however, the stem **5** of the cylinder valve **4** and

the guide rod 13 connected with the armature 11 constitute a one-piece component and further, the opening gas spring OS and the closing gas spring CS are coupled with a common piston 21 which is guided in a common cylinder 22. The electromagnetic actuator 8 seals off the upper opening of the cylinder 22 so that by charging the chambers of the cylinders 7 and 17 with pressure by means of the separate lines 7.1 and 17.1 provided with respective control valves 7.2 and 17.2, the resetting forces required for the back-and-forth motion of the armature 11 may be generated.

The embodiment illustrated in FIG. 3 shows a variant of the FIG. 2 construction. The valve stem 5 of the cylinder valve 4 and the guide rod 13 of the armature 11 are separate as in FIG. 1 and are therefore movable independently from one another. Their free ends projecting into the cylinder 22 are provided with respective pistons 6 and 16 in such a manner that between the pistons 6 and 17 an intermediate space 23 (separate work chamber) is maintained.

The cylinders 7 and 17 are, as in the earlier-described embodiment of FIG. 1, connected with a pressure supply source 18 so that, by virtue of controlling the valves 7.2 and 17.2, a suitable pressure may be adapted in the chambers of the cylinders 7 and 17 to the operational requirements.

The intermediate space 23 has its own pressure line 23.1 which is provided with a controllable valve 23.2 and which leads to a pressure supply source 24. The valve 23.2 is a three-way valve so that by suitably setting the valve 23.2, pressurized medium may be introduced into and removed from the intermediate space 23.

The line 23.1 is so arranged that it opens into the intermediate chamber 23 when the cylinder valve 4 is in its closed position, that is, the armature 11 lies against the closing magnet 9. If, in such a position of the armature 11, the intermediate space 23 is charged with a pressure that is greater than the pressure in the chamber of the cylinder 7 then, corresponding to the duration of such pressurizing, the cylinder valve 4 is opened independently from its actuation by electromagnetic forces generated by the electromagnetic actuator 8. The mass flow of the pressure medium which is preferably also a gas, determines essentially the magnitude of the opening stroke to be effected for the cylinder valve 4. In practice, however, only a small opening stroke is assigned to the cylinder valve 4 as urged by the pressure in the intermediate space 23 in order to provide only a slight valve opening at the beginning of the suction stroke to thus ensure a higher flow velocity of the fresh combustion gas flowing into the combustion chamber of the cylinder. Thereafter, by an energization of the opening magnet 10, the electromagnetic actuator 8 moves the valve 4 into the fully open position, as the armature 11 executes its full stroke toward the electromagnet 10. The control of the valve 23.2 is freely selectable so that an arbitrary adaptation to the operational requirements is possible for the control device 19 forming part of the engine control system.

In the embodiments according to FIGS. 2 and 3, the passage for the guide rod through the electromagnet 10 and the passage for the stem 5 of the cylinder valve 4 have to be provided with a suitable seal.

Reverting to FIG. 1, to determine the position of the armature 11 in relation to the respective pole faces 12 of the electromagnets 9 and 10, at least in one of the pressure chambers, for example, in the cylinder 7, a pressure sensor 26 is disposed whose signal conductor is connected to the control device 19. By monitoring the pressure or its change over time, a prediction concerning the position of the armature may be made as it approaches the pole face 12 of the respective capturing electromagnet. Such a signal may be utilized in the control of the current supply for the capturing electromagnet. It is also feasible to provide a similar pressure sensor in the cylinder 17 as well, so that from the superposition of the two signals (increase of pressure in one cylinder and a corresponding pressure drop in the other cylinder) the reliability of the indication of the armature position may be increased.

It is a further advantage of such pressure sensors that the pressure detected in each instance for the two cylinders 7 and 17 may also be used for controlling the valves 7.2 and 17.2 if, for example, in case of an increasing or decreasing rpm the pressure should be accordingly increased and reduced, respectively.

It will be understood that the above description of the present invention is susceptible to various modifications, changes and adaptations, and the same are intended to be comprehended within the meaning and range of equivalents of the appended claims.

What is claimed is:

1. A combination of a cylinder valve of an internal-combustion engine with an electromagnetic assembly for operating said valve; said assembly comprising

(a) an electromagnetic actuator including
 (1) first and second electromagnets supported in a spaced relationship with respect to one another;
 (2) an armature movable between said first and second electromagnets; and
 (3) coupling means for operatively connecting said armature with said valve for moving said valve between open and closed positions by said armature;
(b) a first control device for energizing said electromagnets to cause a motion of said valve by electromagnetic forces generated by said first and second electromagnets;
(c) first and second pressurizable resetting gas springs operatively coupled to said armature to oppose motions thereof caused by said electromagnetic forces generated by said first and second electromagnets, respectively;
(d) a pressure supply means for supplying pressure to said first and second resetting gas springs;
(e) a second control device for controlling the pressure supplied to said first and second resetting gas springs; and
(f) a pressure sensor disposed in at least one of said first and second resetting gas springs and connected to said first control device for detecting pressures, representing positions of said armature, in said one resetting gas spring.

2. The combination as defined in claim 1, wherein said first and second resetting gas springs are situated externally of said electromagnetic actuator.

3. The combination as defined in claim 1, wherein one of said first and second resetting gas springs is a closing spring urging said valve into said closed position and the other of said first and second resetting gas springs is an opening spring urging said valve into said open position; said closing spring being connected with said valve and said opening spring being connected with said armature.

4. The combination as defined in claim 3, wherein said armature and said opening spring are movable separately from said valve and said closing spring.

* * * * *

US006340007B2

(12) United States Patent
Di Lieto et al.

(10) Patent No.: **US 6,340,007 B2**
(45) Date of Patent: **Jan. 22, 2002**

(54) **METHOD FOR ESTIMATING THE END-OF-STROKE POSITIONS OF MOVING MEMBERS OF ELECTROMAGNETIC ACTUATORS FOR THE ACTUATION OF INTAKE AND EXHAUST VALVES IN INTERNAL COMBUSTION ENGINES**

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(73) Assignee: **Magneti Marelli S.p.A.**, Milan (IT)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **09/739,799**

(22) Filed: **Dec. 20, 2000**

(30) **Foreign Application Priority Data**

Dec. 23, 1999 (IT) BO99A0710

(51) **Int. Cl.**[7] **F01L 9/04**

(52) **U.S. Cl.** **123/90.11**; 251/129.01; 251/129.15

(58) **Field of Search** 123/90.11; 251/129.01, 251/129.05, 129.15, 129.16

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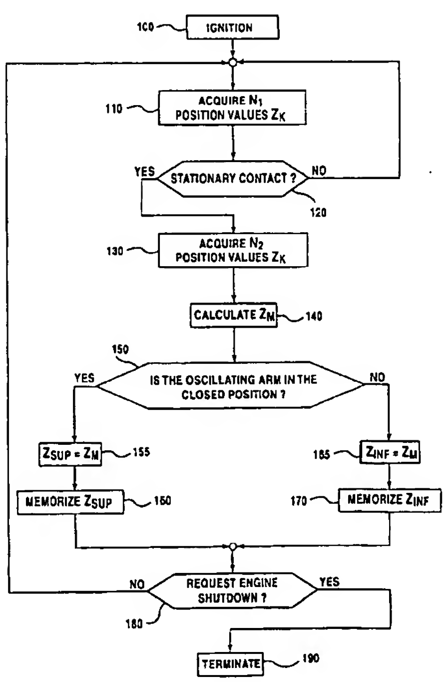
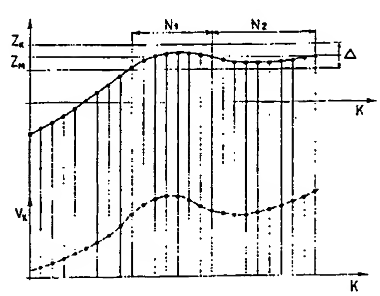
(74) *Attorney, Agent, or Firm*—Arent Fox Kintner Plotkin & Kahn, PLLC

(57) **ABSTRACT**

A method for estimating the end-of-stroke positions of moving members of electromagnetic actuators for the actuation of intake and exhaust valves in internal combustion engines in which an actuator is coupled to a respective intake or exhaust valve and comprises a moving member actuated magnetically in order to control the movement of the valve, a sensor supplying a position signal representative of a current position of this moving member and a first and a second electromagnet disposed on opposite sides of the moving member, wherein this moving member can move between a first end-of-stroke position in which it is disposed in contact with the first electromagnet and a second end-of-stroke position in which it is disposed in contact with the second electromagnet. The method comprises the stages of checking whether the condition of stationary contact of the moving member exists and determining a magnitude correlated with this current position, if the stationary condition is verified.

6 Claims, 4 Drawing Sheets





See figs 2a/2b

01/21/2004, EAST Version: 1.4.1

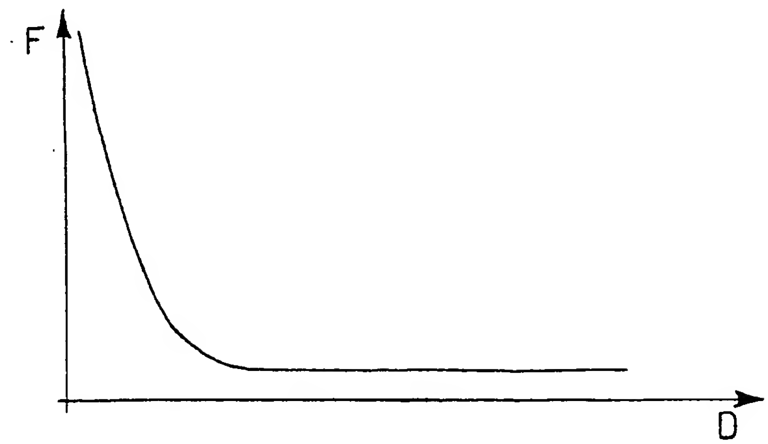

F
D

Fig.1

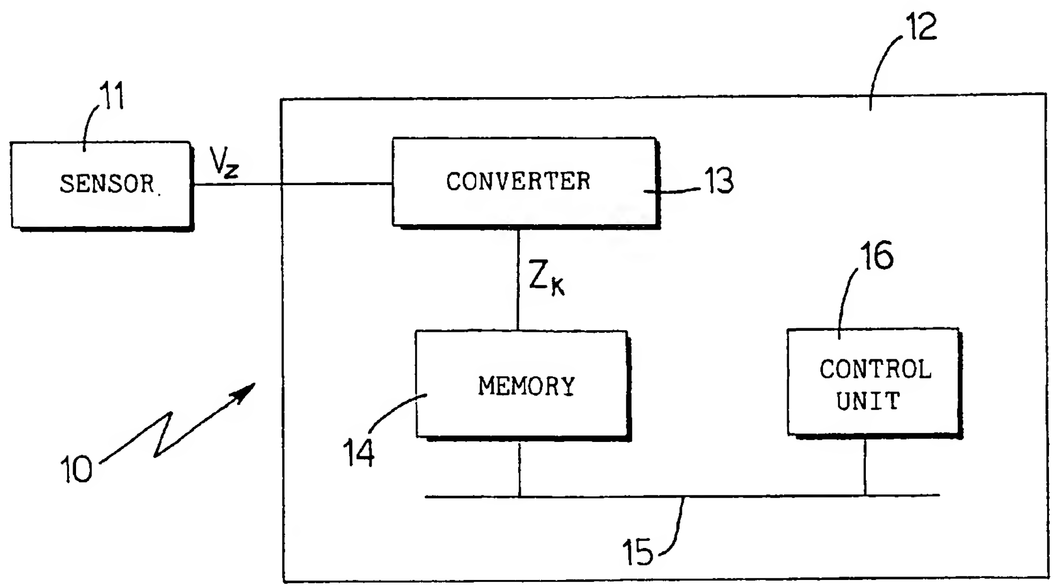

12
11
SENSOR
V_z
CONVERTER
13
Z_K
16
MEMORY
CONTROL UNIT
10
14
15

Fig.3

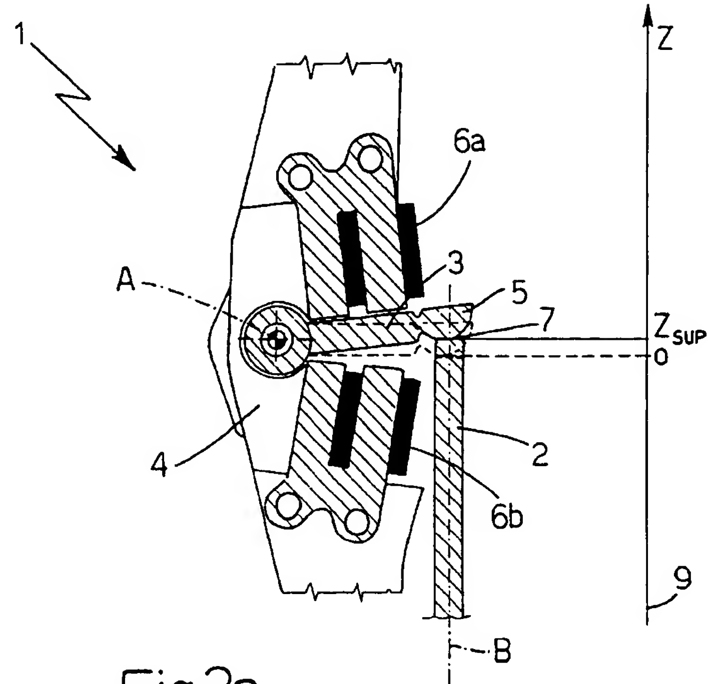


Fig.2a

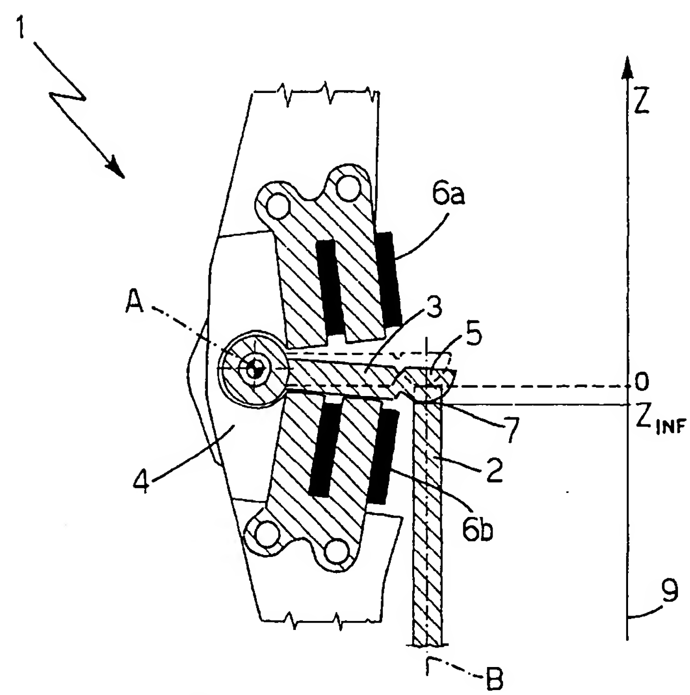
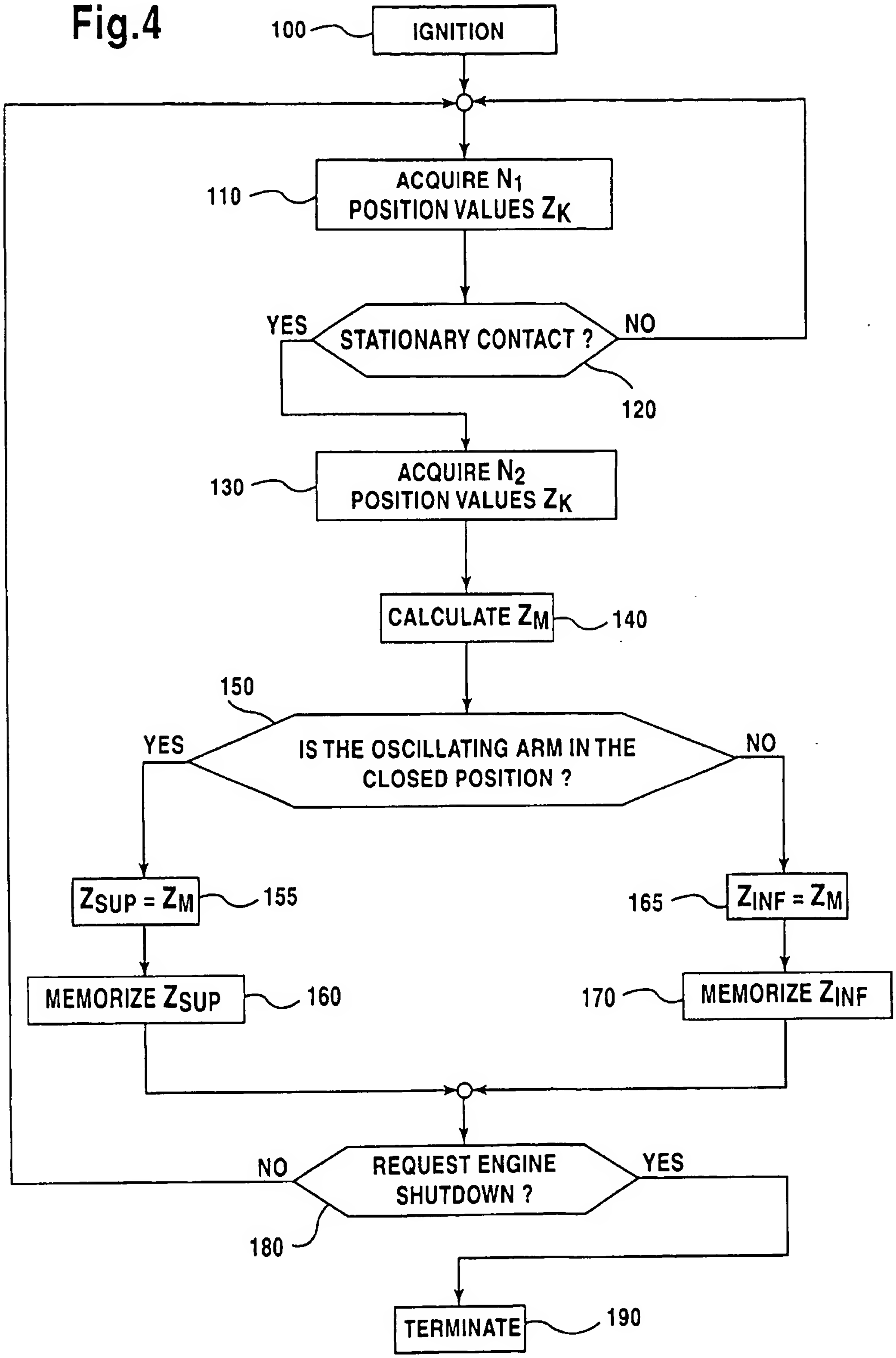


Fig.2b

Fig.4



100
IGNITION
110
ACQUIRE N_1 POSITION VALUES Z_K
YES
STATIONARY CONTACT ?
NO
120
130
ACQUIRE N_2 POSITION VALUES Z_K
CALCULATE Z_M
140
150
YES
IS THE OSCILLATING ARM IN THE CLOSED POSITION ?
NO
$Z_{SUP} = Z_M$
155
165
$Z_{INF} = Z_M$
MEMORIZE Z_{SUP}
160
170
MEMORIZE Z_{INF}
NO
REQUEST ENGINE SHUTDOWN ?
YES
180
TERMINATE
190

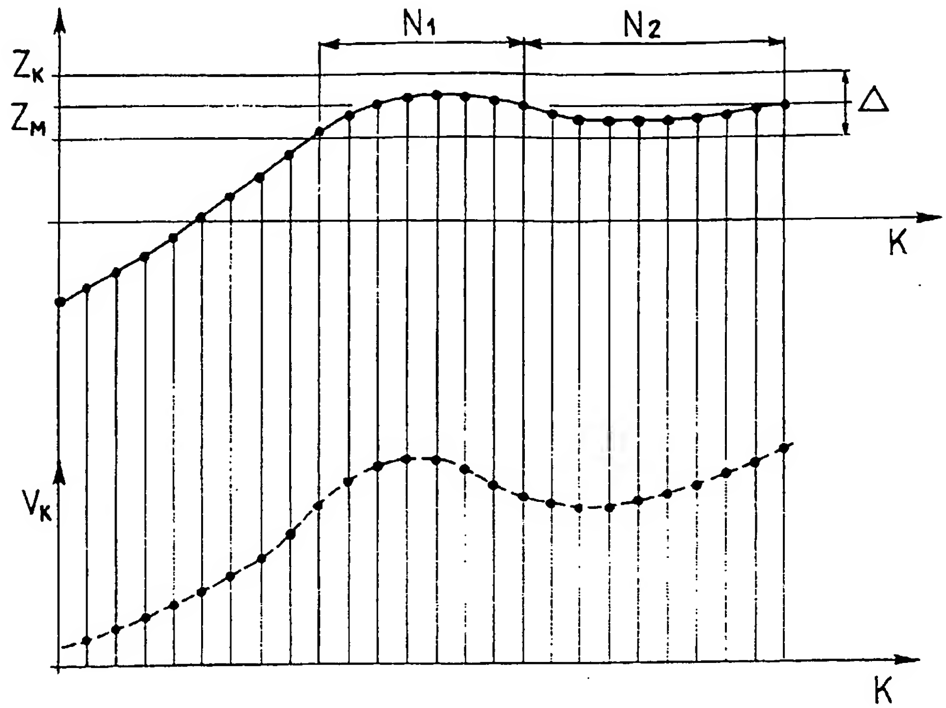


Fig.5

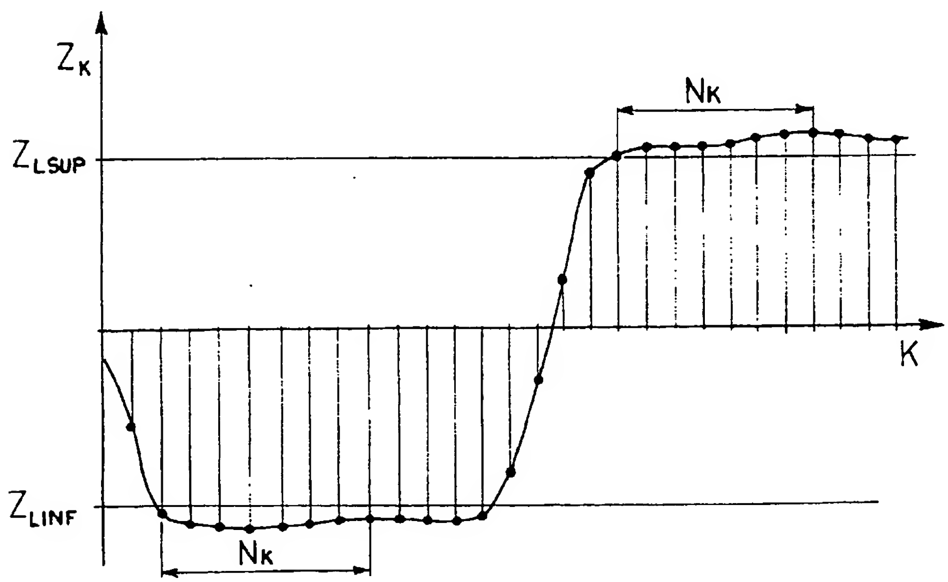


Fig.6

METHOD FOR ESTIMATING THE END-OF-STROKE POSITIONS OF MOVING MEMBERS OF ELECTROMAGNETIC ACTUATORS FOR THE ACTUATION OF INTAKE AND EXHAUST VALVES IN INTERNAL COMBUSTION ENGINES

The present invention relates to a method for estimating the end-of-stroke positions of moving members of electromagnetic actuators for the actuation of intake and exhaust valves in internal combustion engines.

BACKGROUND OF THE INVENTION

As is known, drive units are currently being tested in which the actuation of the intake and exhaust valves is managed by using actuators of electromagnetic type, which replace purely mechanical distribution systems (camshafts).

These actuators in particular comprise a pair of electromagnets disposed on opposite sides of a moving ferromagnetic member connected to a respective intake or exhaust valve and held in a rest position by elastic members (for instance a spring and/or a torsion bar). The moving ferromagnetic member is actuated by applying a force generated by the electromagnets in order to be brought into contact alternatively with one or other of these electromagnets, so as to move the corresponding valve between a closed position and a position of maximum opening according to desired timings and trajectories. In this way, it is possible to actuate the valves according to optimum lift profiles in any operating condition of the engine, thereby substantially improving overall performance.

Obtaining an actual increase in the efficiency of the engine is conditioned, however, by the precision of the systems and methods used for the control of the actuators. In order, in particular, accurately to control the force transmitted by the electromagnets to the moving member and thus the movement of the valve, it is indispensable to have an accurate measurement of the distances intervening between the moving member and the polar heads of one or the other electromagnet. As shown by way of example in FIG. **1**, the force F that an electromagnet is able to transmit to the moving member depends, the current absorption being equal, in a highly non-linear manner on the distance D between the polar head of the electromagnet and the moving member. An error, even of a few microns, in the measurement of the distance D, in particular for low values of the latter, may therefore compromise the efficiency of the control and thus entail a substantial deterioration of the performance of the engine.

This is a serious drawback, given that internal combustion engines are subject, during their use, to substantial temperature variations which cause expansions and/or contractions of the materials, especially of the metal parts. Consequently, even the polar heads of the electromagnets may expand or contract as a function of temperature, thereby affecting the measurement of the distances between these electromagnets and the moving member.

SUMMARY OF THE INVENTION

The object of the present invention is to provide a method for estimating the end-of-stroke positions of the moving member which makes it possible to remedy the above-mentioned drawbacks and, in particular, makes it possible to reduce the overall consumption of electrical power.

The present invention therefore relates to a method for estimating the end-of-stroke positions of moving members of electromagnetic actuators for the actuation of intake and exhaust valves in internal combustion engines, in which an actuator is coupled to a respective intake or exhaust valve and comprises a moving member actuated magnetically in order to control the movement of the valve, a sensor supplying a position signal representative of a current position of this moving member and a first and a second electromagnet disposed on opposite sides of this moving member, wherein this moving member can move between a first end-of-stroke position in which it is disposed in contact with the first electromagnet and a second end-of-stroke position in which it is disposed in contact with the second electromagnet, which method is characterised in that it comprises the stages of:

a) checking whether the moving member is in a condition of stationary contact; and

b) determining a magnitude correlated with this current position, if the condition of stationary contact is verified.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention is set out in further detail below with reference to an embodiment thereof, given purely by way of non-limiting example and made with reference to the accompanying drawings, in which:

FIG. **1** is a graph relating to an electromagnetic actuator;

FIGS. **2***a* and **2***b* are lateral elevations, partly in cross-section, of an electromagnetic actuator and the corresponding intake or exhaust valve in a first and a second end-of-stroke position respectively;

FIG. **3** is a simplified block diagram relating to the control method of the present invention;

FIG. **4** is a flow diagram relating to the present method; and

FIGS. **5** and **6** are graphs relating to curves of magnitudes of the present method.

DETAILED DESCRIPTION OF THE INVENTION

In FIGS. **2***a* and **2***b*, an electromagnetic actuator **1** is coupled to an intake or exhaust valve **2** of an internal combustion engine. The actuator **1** comprises an oscillating arm **3** of ferromagnetic material having a first end hinged on a fixed support **4** so as to be able to rotate about a horizontal axis of rotation A perpendicular to a longitudinal axis B of the valve **2**. A second end **5** of the oscillating arm **3** cooperates in contact, moreover, with an upper end of the valve **2** so as to impose an alternating movement in a direction parallel to the longitudinal axis B on this valve **2**.

The actuator **1** comprises a closing electromagnet **6***a* and an opening electromagnet **6***b* disposed on opposite sides of the body of the oscillating arm **3**, in order to be able to act on command, in sequence or simultaneously, by exerting a net force on the oscillating arm **3** in order to cause it to rotate about the axis of rotation A.

Moreover, a first and second elastic member, for instance a spring and a torsion bar, not shown for the sake of simplicity, act so that the oscillating arm **3** is maintained in a rest position in which it is equidistant from the polar heads of the closing and opening electromagnets **6***a* and **6***b* respectively.

FIGS. **2***a* and **2***b* also show a reference axis **9**, oriented parallel to the longitudinal axis B of the valve, on which a coordinate of a point representative of the position of the

oscillating arm 3 is shown (for instance the point of a lower edge 7 of the second end 2 which, at any moment, is located at the longitudinal axis B). In the following description, "position Z" is used to refer to this coordinate. Given that the end 5 normally acts in abutment against the upper end of the valve 2, the current position Z is also representative of the position of the valve 2.

In FIG. 2*a*, in particular, the oscillating arm 3 is shown in a first end-of-stroke position or closed position, corresponding to a closed position value Z_{SUP} on the reference axis 9. When in this position, the oscillating arm 3 is disposed in contact with the polar head of the closing electromagnet 6*a* and therefore the position of the latter is represented by the closed position value Z_{SUP}. It will be appreciated that, in this situation, the second end 5 of the oscillating arm 3 may be detached from the upper end of the valve 2 since this valve 2 reaches a limit position Z_{LIM}, in which it is kept closed. Even during a phase of detachment, however, the current position Z is representative of the actual position of the valve 2: values of the current position Z greater than the limit position Z_{LIM} show that the valve 2 is closed and is exactly in the limit position Z_{LIM}.

In FIG. 2*b*, however, the oscillating arm 3 is shown in a second end-of-stroke position, i.e. a position of maximum opening, in which it is disposed in contact with the polar head of the opening electromagnet 6*b*. This position of maximum opening, which corresponds to a maximum opening value Z_{INF} on the reference axis 9, is therefore also representative of the position of the polar head of the closing electromagnet 6*a* and also coincides with the position of maximum opening of the valve 2.

In both FIG. 2*a* and FIG. 2*b*, moreover, the oscillating arm 3 is shown, in dashed lines, in the rest position, which is taken as the origin of the reference axis 9.

As shown in FIG. 3, in a control system 10 of the actuator 1, a position sensor 11, of known type, supplies a position signal V_Z representative of the current position Z of the oscillating arm 3 to an electronic control unit 12. The electronic control unit 12 is provided with a converter 13 which receives as input the position signal V_Z, samples it at a predetermined sampling frequency and, in a manner known per se, supplies as output position values Z_K correlated with sampling values V_K assumed by the position signal V_Z at each sampling moment K.

The position values Z_K acquired are stored in a memory 14, which, by means of a bus 15, is connected to a control unit 16 adapted to carry out procedures for the control of the operation of the engine. Moreover, the closed position value Z_{SUP} and the maximum opening position value Z_{INF} are also stored in the memory 14.

With reference to FIG. 4, the method of the present invention provides that, following ignition of the engine (block 100), a first number N_1, for instance 50, of position values Z_K (block 110) is initially acquired.

Subsequently, a test is carried out to check whether there is a condition of stationary contact of the valve 2, which exists when the oscillating arm 3 is held in the closed position Z_{SUP} or the position of maximum opening Z_{INF} (block 120). In particular, it is checked whether the difference between the maximum position value Z_{KMAX} and the minimum position value Z_{KMIN} among the N_1 values of position Z_K acquired is smaller than a predetermined range threshold Δ.

If the outcome of the test is negative (output NO from the block 120), a new set of N_1 values of position Z_K is again acquired (block 110). If, however, the stationary conditions are verified (output YES from the block 120), a second number N_2, for instance 200, of position values Z_K are acquired (block 130), of which a mean value Z_M (block 140) is then calculated according to the equation:

$$Z_M = \frac{1}{N_2}\sum_{K=1}^{2} Z_K \qquad (1)$$

It is then checked whether the oscillating arm 3 is in the closed position, verifying whether the mean value Z_M is positive (block 150). If so (output YES from the block 150), i.e. if the oscillating arm 3 is in contact with the polar head of the closing electromagnet 6*a*, the closed position Z_{SUP} is set to Z_M (block 155) and then memorised (block 160). If the mean value Z_M is negative (output NO from the block 150) and therefore the oscillating arm 3 is in the position of maximum opening Z_{INF}, in contact with the polar head of the opening electromagnet 6*b*, the position of maximum opening Z_{INF} is set to the mean value Z_M (block 165) and memorised (block 170).

Subsequently, it is checked whether stoppage of the engine has been requested (block 180). If so (output YES from the block 180), the estimation procedure is terminated (block 190); otherwise (output NO from the block 180), a set of N_1 values of position Z_K is again acquired (block 110).

FIG. 5 shows, by way of example, a curve of the position values Z_K (represented by points connected by a continuous line) and of the corresponding sampling values V_K (shown by points connected by dashed lines), as a function of the generic moment of sampling K; the first and the second number N_1, N_2 of position values acquired and the range threshold Δ are also shown.

In practice, the end-of-stroke positions of the oscillating arm 3 (closed position and position of maximum opening) are estimated when it is recognised that the oscillating arm 3 is substantially stationary, i.e. when its actual position Z has not changed significantly for a time sufficient to acquire the first number of position values Z_K. In this case, further position values Z_K are acquired and their mean value Z_M is calculated. In particular, the second number N_2 of position values Z_K acquired must be high enough so that any disturbances, for instance noise present in the position signal V_Z, has no impact on the calculation of the mean value Z_M. The mean value Z_M is then memorised as a new closed position value Z_{SUP}, if positive, or as a maximum opening position value Z_{INF}, if negative. Given that, in each engine cycle, the valve 2 and therefore the oscillating arm 3 stop at least once in the closed position and in the position of maximum opening, both the values of the closed position Z_{SUP} and of the position of maximum opening Z_{INF} can be rapidly updated in succession. Moreover, the estimate of the end-of-stroke positions is repeated each time that the condition of stationary contact is verified, until the stoppage of the engine is requested.

The estimation method as described has the following advantages.

In the first place, it is possible to update the estimate of the end-of-stroke positions in real time, given that the estimation procedure is carried out each time that stationary contact conditions are detected. Consequently, a precise estimate of the positions of the polar heads of the closing and opening electromagnets 6*a* and 6*b* is also supplied in real time.

It is therefore possible to obtain a correct measurement of the distance intervening between the polar heads of the

electromagnets and the oscillating arm, irrespective of variations due to heat expansion.

In particular, the method of the present invention may be advantageously used for instance in the case of the method for the control of electromagnetic actuators as disclosed in Italian Patent Application B099A000594 of Nov. 5, 1999 filed in the name of the applicants.

This Patent Application relates to the control of an electromagnetic actuator, substantially of the type of the actuator 1 described in FIGS. 2*a* and 2*b*, to which reference will continue to be made. According to the method disclosed in the above-mentioned Application, a feedback control of the actual position Z and of an actual velocity V of the valve **2** is carried out, using, as the control variable, the net force applied by means of the opening and closing electromagnets *6a* and *6b* to the oscillating arm **3**, which actuates this valve **2**. For this purpose, by means of a model based on a dynamic system, an objective force value F_O to be exerted on the oscillating arm **3** is calculated as a function of an actual position, an actual velocity, a reference position and a reference velocity of the valve. The dynamic system is in particular described by the following matricial equation:

$$\begin{bmatrix} \dot{Z} \\ \dot{V} \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ K/M & B/M \end{bmatrix} \begin{bmatrix} Z \\ V \end{bmatrix} + \begin{bmatrix} 0 \\ 1/M \end{bmatrix} F \tag{2}$$

in which Z and V are the time derivatives of the actual position Z and of the actual velocity V respectively, F is the net force exerted on the oscillating arm **3**, K is an elastic constant, B is a viscous constant and M is an equivalent total mass. In particular, the net force F and the actual position Z respectively represent an input and an output of the dynamic system.

Moreover, the objective force value F_0 is calculated by the equation:

$$F_O=(N_1Z_R+N_2V_R)-(K_1Z+K_2V) \tag{3}$$

in which N_1, N_2, K_1 and K_2 are gains that may be calculated by applying well-known robust control techniques to the dynamic system represented by equation (2).

Subsequently, the current values to be supplied to the closing and opening electromagnets *6a* and *6b* are calculated so that the net force exerted on the oscillating arm **3** has a value equal to the objective force value F_O.

Clearly, given that the net force applied, as discussed above, is highly dependent on the actual distance intervening between the oscillating arm **3** and the polar heads of the closing and opening electromagnets *6a* and *6b*, the use of the present estimation method in the case described in the above-mentioned Patent Application makes it possible substantially to improve the accuracy and reliability of the control.

It will be appreciated that modifications and variations may be made to the method as described, without departing from the scope of the present invention.

In particular, the condition of stationary contact of the oscillating arm **3** (FIG. **4**, block **120**) could be evaluated in a different way. For instance, it is possible to check whether a minimum number N. of consecutive position values Z_K are alternately greater than an upper limit position Z_{LSUP} (oscillating arm **3** in the closed position) or lower than a lower limit position Z_{LINF} (oscillating arm **3** in the position of maximum opening) as shown in FIG. **6**. As an alternative, it is possible to verify whether the velocity of the oscillating

arm is below a predetermined threshold, or whether the currents supplied to the closing or opening electromagnets *6a* and *6b* continue to be substantially constant.

What is claimed is:

1. A method for estimating the end-of-stroke positions of moving members of electromagnetic actuators for the actuation of intake and exhaust valves in internal combustion engines, in which an actuator **(1)** is coupled to a respective intake or exhaust valve **(2)** and comprises a moving member **(3)** actuated magnetically in order to control the movement of the valve **(2)**, a sensor **(11)** supplying a position signal (V_Z) representative of a current position (Z) of this moving member **(3)** and a first and a second electromagnet **(*6a*, *6b*)** disposed on opposite sides of this moving member **(3)**, wherein this moving member **(3)** can move between a first end-of-stroke position (Z_{SUP}) in which it is disposed in contact with the first electromagnet **(*6a*)** and a second end-of-stroke position (Z_{INF}) in which it is disposed in contact with the second electromagnet **(*6b*)**, which method is characterised in that it comprises the stages of:

a) checking whether a condition of stationary contact of the moving member **(3)** exists **(110, 120)**; and

b) determining a value (Z_M) correlated with this current position (Z) **(130, 140)**, if the condition of stationary contact is verified.

2. A method as claimed in claim **1**, characterized in that the stage a) of checking whether the condition of stationary contact exists comprises the stage of:

a1) acquiring a first number (N_1) of position values (Z_K) correlated with sampling values (V_K) of the position signal (V_Z) at predetermined sampling moments **(110)**.

3. A method as claimed in claim **2**, characterised in that the stage a) of checking whether the condition of stationary contact exists further comprises the stage of:

a2) checking whether the difference between a maximum position value (Z_{KMAX}) and a minimum position value (Z_{KMIN}) is lower than a range threshold (D).

4. A method as claimed in claim **2**, characterised in that the stage a) of checking whether the condition of stationary contact exists further comprises the stage of:

a3) checking whether the position values (Z_K) acquired are greater than an upper limit position (Z_{LSUP}),

a4) checking whether the position values (Z_K) acquired are lower than a lower limit position (Z_{LSUP}).

5. A method as claimed in claim **1**, characterised in that the stage b) of determining a value (Z_M) comprises the stages of:

b1) acquiring a second number (N2) of position values (Z_K)correlated with sampling values (V_K) of the position signal (V_Z) at predetermined sampling moments **(130)**; and

b2) calculating a mean value (Z_M) of the position values (Z_K)acquired **(140)**.

6. A method as claimed in claim **5**, characterised in that the stage b2) of calculating the mean value (Z_M) is followed by the stages of:

b3) determining whether the moving member **(3)** is in the first end-of-stroke position **(150)**; and

b4) determining whether the moving member **(3)** is in the second end-of-stroke position **(150)**.

* * * * *

L Number	Hits	Search Text	DB	Time stamp
1	148962	electro adj magnet or electromagnet	USPAT; EPO; JPO; DERWENT	2004/01/21 10:13
2	228521	solenoids	USPAT; EPO; JPO; DERWENT	2004/01/21 10:14
3	367127	(electro adj magnet or electromagnet) or solenoids	USPAT; EPO; JPO; DERWENT	2004/01/21 10:14
4	1354882	valve or 251.clas.	USPAT; EPO; JPO; DERWENT	2004/01/21 10:15
5	156617	((electro adj magnet or electromagnet) or solenoids) and (valve or 251.clas.)	USPAT; EPO; JPO; DERWENT	2004/01/21 10:17
6	5476	251/129.09-129.2.ccls.	USPAT; EPO; JPO; DERWENT	2004/01/21 10:17
7	3275	((electro adj magnet or electromagnet) or solenoids) and 251/129.09-129.2.ccls.	USPAT; EPO; JPO; DERWENT	2004/01/21 10:19
8	46944	oppos?	USPAT; EPO; JPO; DERWENT	2004/01/21 10:19
9	83	(((electro adj magnet or electromagnet) or solenoids) and 251/129.09-129.2.ccls.) and oppos?	USPAT; EPO; JPO; DERWENT	2004/01/21 10:26